TWO SECTIONS

AUG 6 1930

SECTION 1

S·A·E JOURNAL

AUGUST 1930

AN HYDRAULIC SHOCK ABSORBER • • **does its work by forcing oil out of a cylinder through a small orifice. The friction of the oil passing through this orifice provides the resistance necessary to hold the car. That the cylinder should be full of oil is obviously necessary if work is to be done.**

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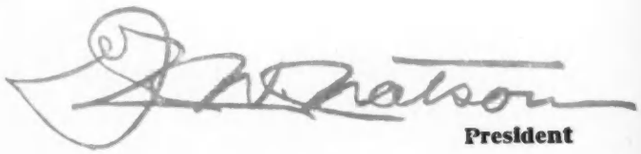
That is just one of the reasons why these new Watson Supercharging Hydraulics do the work.

JOHN WARREN WATSON COMPANY
Philadelphia, Pa.



President

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The purpose of meetings of the Society is largely to provide a forum for the presentation of straightforward and frank discussion. Discussion of this kind is encouraged. However, owing to the nature of the Society as an organization, it cannot be responsible for statements or opinions advanced in papers or in discussions at its meetings. The Constitution of the Society has long contained a provision to this effect.

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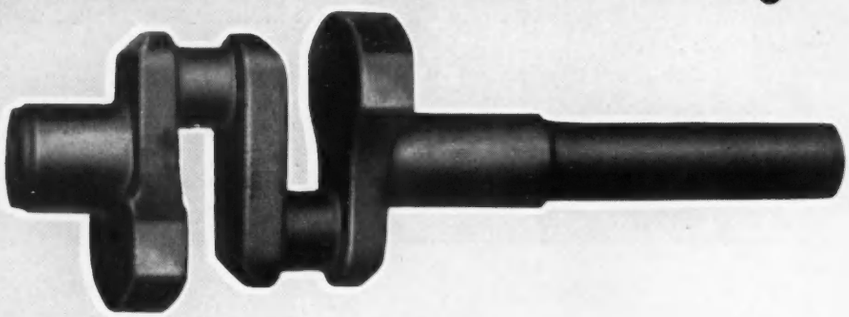
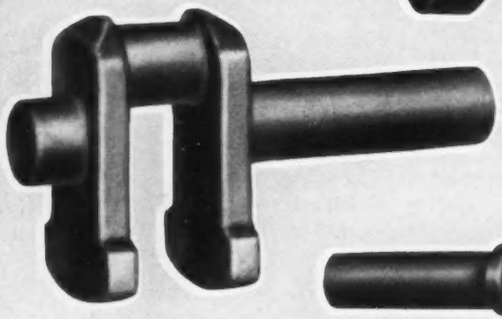
. . . counterweights forged in one piece.





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THE Crankshaft may properly be regarded as the most important single part of an engine. Quality may sometimes safely be sacrificed in other parts, in the interests of economy, but in the Crankshaft nothing but the *best* should be tolerated!

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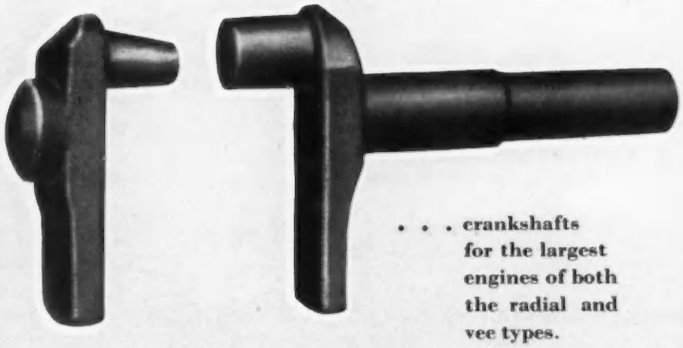

. . . crankshafts for the largest engines of both the radial and vee types.

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"GUARANTEED FORGINGS"

S. A. E. JOURNAL

Vol. XXVII August, 1930 No. 2

Coming National Meetings

Joint Aeronautic Meeting August 26 to 28 in Chicago with Aeronautical Chamber—Production, Transportation and Annual Meeting Plans

IN FORMULATING the program printed on p. 133 of this issue of THE JOURNAL for the three-day Aeronautic Meeting, to be held jointly with the Aeronautical Chamber of Commerce of America at the Palmer House, Chicago, the last week in August, the intention was to present papers that will bring out discussion on the main problems now confronting the aircraft industry from which will come something tangible in the way of their solution.

Only forenoon and evening sessions are to be held, the afternoons being left open to afford opportunity to attend the National Air Races at the Curtiss-Wright-Reynolds Field under the auspices of the National Air Race Association.

On Tuesday morning, Aug. 26, at the Engine Session, under the Chairmanship of J. H. Geisse, E. D. Herrick will talk on the Manufacturing Costs of Aircraft Engines and H. M. Crane will give his views on Comparative Data on Powerplants for Motor-Cars and Aircraft.

B. Russell Shaw will guide the Airplane Design Session on the evening of the 26th, when J. W. Crowley, Jr., will talk on Flight Research and William B. Stout on the Possibilities of Radical Airplane Design. At another Airplane Design Session on Wednesday morning, Aug. 27, of which Mac Short will be Chairman, Charles Ward Hall and A. A. Gassner will give their ideas in separate papers on the subject of Weight Saving by Structural Efficiency.

The evening session on Wednesday will take the form of a symposium on Spinning Characteristics and Control, at which Paul E. Hovgard, Garland P. Peed, Jr., Temple Joyce, H. A. Sutton, Lieut. Carl B. Harper and Fred E. Weick will present their views on all phases of spins, their causes and prevention. This session will be under the Chairmanship of Dr. George W. Lewis.

Another symposium, on Aircraft Fuels, will comprise the morning session on Thursday, Aug. 28. It is believed that the facts presented at this session will be of great value in reaching a better understanding between aircraft-fuel consumers and producers, as the following authorities will give their views with this in mind: E. E. Aldrin, C. S. Fliedner, S. D. Heron, C. M. Larson and W. A. Parkins.

The meeting will close with an Aircraft Banquet, at which William B. Stout will preside as Toastmaster. The name of the speaker of the evening, who will be a National figure in the aeronautic industry, will be announced later.

Two Meetings in October

Papers are being written and arrangements made for the Production Meeting to be held Oct. 7 and 8 at the Book-Cadillac Hotel in Detroit. The activities in conjunction with this meeting include three technical sessions, a Production Dinner to be held the evening of the second day, and one or more inspection visits to local manufacturing plants. The Detroit Section is arranging the program for the Dinner, under the direction of Section Chairman Phil Kent. J. Geschelin, Chairman of the Meetings Committee of the Production Activity Committee, has charge of arranging the program for the technical sessions. In its final form this will be published in THE JOURNAL for September. Two of the papers, for the Production Economics Session, have been preprinted and many others will be available in preprint form before the meeting.

Other subjects to be covered by papers in preparation are: New Developments in Machining Aluminum and Its Alloys, The Use of Cemented Tungsten-Carbide Tools, Design Elements and Economic Conditions of Material-Handling Systems, and Wear Allowances and Tolerances on Gages.

This year's Transportation Meeting is to be held in Pittsburgh, at the William Penn Hotel, on Oct. 22, 23 and 24. There are to be forenoon and afternoon sessions each day, a Transportation Dinner the evening of the second day, and several visits to local fleet-operating plants. A full program of this meeting will be published in the October issue of THE JOURNAL.

Annual Dinner, Jan. 8

Thursday evening, Jan. 8, is the time selected for the 1931 Annual Dinner. The place is the Pennsylvania Hotel, New York City. As usual, the Dinner will be held in the week of the New York Automobile Show, which next winter is scheduled for Jan. 3 to 10. The program that has been arranged for the Dinner assures that it will start the new year auspiciously. The toastmaster and speakers will be announced later. Results of the election of officers of the Society for 1931 will be reported at the Dinner by the tellers of election.

Annual Meeting, Jan. 19 to 23

Plans are well under way for the 1931 Annual Meeting, which is to be held in Detroit as usual during the week of the Detroit Automobile Show. The program for this meeting, which is to start on Monday and extend through five days, is now rapidly taking shape.

The Transportation and Maintenance, the Aircraft, the Aircraft-Engine and the Passenger-Car-Body Activities plan to sponsor one session each, while the Diesel-Engine Activity has requested that a whole day be set aside for its sessions. The Meetings Committee has consented to the request of the Cooperative Fuel-Research Steering Committee that a session be allotted for holding a symposium on detonation.

Following a series of informal meetings, the Passenger-Car Activity Committee met recently in Detroit and decided that it should sponsor four sessions in addition to the Detonation Symposium. As tentatively planned, these will be an Engine, a Chassis, a Fuel and Lubrication, and a Research Session.

The 1931 Annual Meeting will mark

(*Concluded on p.* 133)

Members Approve Changes in Dues

Constitutional Amendments Combining Society and Section Dues Carried by Mail Vote and Effective October 1

AMENDMENTS to Sections C 20, C 21, C 21a and C 52 of the Constitution of the Society, making changes in the annual dues and student Enrollment, were carried by letter-ballot dated June 6, 1930. Of the 3500 ballots mailed to voting members, the report of the tellers shows that a total of 1106 votes were cast, and of these 11 were defective. The tellers were William Harrigan, Austin M. Wolf and F. L. Faurote. The valid votes were divided as follows:

Section	Favoring Adoption of Proposed Amendments	Opposing Adoption of Proposed Amendments
C 20	916	179
C 21	731	364
C 21a	824	271
C 52	935	160

Section C 20, as amended, makes no changes in initiation fees except to omit reference to Student Enrollment.

Section C 21, as amended, changes the annual dues for membership in the several grades as follows, the portions in parentheses to be omitted and the figures in italics to prevail:

For Member	($15.00)	*$20.00*
For Associate	(15.00)	*20.00*
For Junior	10.00	10.00
For Service Member	(10.00)	*15.00*
For Foreign Member	(10.00)	*15.00*
For Student Enrollment	(3.00)	
For Departmental Member (For One Affiliate Member Representative	15.00)	None
For each (additional) Affiliate Member Representative	(10.00)	*20.00*

NOTE — Each Affiliate Member Representative would receive the publications of the Society on the same basis as individual members receive them and his name would be included in alphabetical, geographical and company lists of the Society Roster.

Dues Include Section Dues

In the cases of Members, Associate Members and Affiliate Member Representatives the increased dues represent a combination of Society dues and Section dues, both to be collected by the Society. A new section is added to the Constitution, by approval of the amendments, to provide for payment of part of the dues to the Sections, in lieu of Section dues. This new section reads:

C 21a—The Society shall turn over to each Section of the Society a part of the Society dues, which shall be specified in the By-Laws, paid by each Member and Associate and for each Affiliate Member Representative residing in the territory of each Section as defined by the Council.

Meetings Calendar

Chicago Aeronautic—Aug. 26 to 28

In conjunction with the National Air Races

Palmer House

Production—Oct. 7 and 8

Book-Cadillac Hotel, Detroit

Production Dinner, Oct. 8

Transportation—Oct. 22 to 24

William Penn Hotel, Pittsburgh

Transportation Dinner, Oct. 23

Annual Dinner—Jan. 8

Hotel Pennsylvania, New York City

Annual Meeting—Jan. 19-23

Book-Cadillac Hotel, Detroit

By action of the Council, the By-Laws have been amended to accord with this section of the Constitution by the addition of the following section of the By-Laws:

B 31a—Upon payment of the Society dues by each Member and Associate and for each Affiliate Member Representative, there shall be available for the use of the Section in whose territory the Member, Associate or Affiliate Member resides, an amount of $5 which shall be considered as Section dues collected for the Section by the Society. This sum shall be remitted to the Section Treasurer as and if needed to carry on Section activities of approved types. Additional sums may be granted to Sections in the discretion of the Council.

Corresponding changes have been made in the Section Constitution and By-Laws. These were published in full in the January, 1930, issue of the S.A.E. JOURNAL, p. 6, and provide that all members of the Society and Affiliate Member Representatives in good standing residing in the territory of a Section shall be members of the Section, and that those residing outside of a Section territory may become members of a Section upon request.

It is expected that the effect of these provisions will be to increase the membership of the Sections and to provide them with larger funds for carrying on their meetings activities.

Changes in Student Enrollment

The section of the Constitution of the Society relating to Student Enrollment has been amended to read as follows, the words in parentheses being omitted and those in italics inserted:

C 52—Second Paragraph—The Council may further, in its discretion, authorize the enrollment, individually or by group, of persons under 30 years of age, who at the time of application, shall be bona-fide students of a recognized institution of engineering or pursuing (a) *an approved* course of study in (automobile) *automotive* engineering. Student Enrollment (shall) *may* be for (a term of three years, which term shall not be renewable) *the duration of the student's recognized or approved course plus one year thereafter but in no case for longer than five years. The Council shall determine the amount of the annual enrollment fee.* Enrolled Students shall not be members of the Society nor be entitled to vote or hold office.

The Council has voted to fix the Student Enrollment fee at $3 per year for the term at college and $5 for one year after the student leaves college or finishes his engineering course.

Bills for membership dues in accordance with the foregoing amendments will be issued as of Oct. 1 covering dues payable for the fiscal year 1930-1931.

Chicago Aeronautic Meeting Program

Palmer House, Aug. 26 to 28, during the National Air Races

Tuesday, Aug. 26

10 A.M.—ENGINES

J. H. Geisse, *Chairman*
Vice-President, Comet Engine Corp.

Manufacturing Costs of Aircraft Engines—E. D. Herrick, Chief Engineer, Lycoming Mfg. Co.

Comparative Data on Powerplants for Motor-Cars and Aircraft—H. M. Crane, Assistant to President, General Motors Corp.

8 P.M.—AIRPLANE DESIGN

B. Russell Shaw, *Chairman*
President, B. Russell Shaw Co., Inc.

Flight Research—J. W. Crowley, Jr., National Advisory Committee for Aeronautics.

The Possibilities of Radical Airplane Design—William B. Stout, President, Stout Air Services, Inc.

Wednesday, Aug. 27

10 A.M.—AIRPLANE DESIGN

Mac Short, *Chairman*
Vice-President and Chief Engineer, Stearman Aircraft Co.

Weight Saving by Structural Efficiency—Charles Ward Hall, President, Hall-Aluminum Aircraft Corp.

Weight Saving by Structural Efficiency—A. A. Gassner, Chief Engineer, Fokker Aircraft Corp.

8 P.M—SPINNING CHARACTERISTICS AND CONTROL

Dr. George W. Lewis, *Chairman*
Director of Aeronautical Research, National Advisory Committee for Aeronautics.

Symposium

Lieut. Carl Harper, U. S. N.
Paul E. Hovgard, Keystone Aircraft Corp.
Garland P. Peed, Jr., Aeronautical Engineer
Temple Joyce, Berliner-Joyce Aircraft Corp.
H. A. Sutton, Aviation Corp.
Fred E. Weick, National Advisory Committee for Aeronautics

Thursday, Aug. 28

10 A.M.—AIRCRAFT FUELS

Wesley L. Smith, *Chairman*
Superintendent, Eastern Division, National Air Transport, Inc.

Symposium

E. E. Aldrin, Standard Oil Co. of New Jersey
C. S. Fliedner, Bureau of Aeronautics, Navy Department
S. D. Heron, Materiel Division, Air Corps, United States Army
C. M. Larson, Sinclair Refining Co.
W. A. Parkins, Pratt & Whitney Aircraft Co.

7:30 P.M.—AIRCRAFT BANQUET

William B. Stout, *Toastmaster*
Speaker to be announced later

Coming National Meetings

(*Concluded from p.* 131)

an innovation in two respects. A movement is on foot to coordinate, to some extent, National and Section meeting work and to give direction to the Society's literature; that is, for the Activity Committees each to designate two or three topics of current interest and then for an effort to be made to secure papers for both Section and National meetings on those topics over a certain period. This is expected to result in a more definite and valuable contribution to the literature on the subject than could result from a haphazard choice of subjects. The desirability of dealing at a National meeting with a subject that has been presented at Section meetings, especially if there is variation in the material and in its manner of presentation, was also considered.

The Passenger-Car Activity Committee, at its Detroit meeting, went on record unanimously as favoring the practice of scheduling each paper as to time of beginning and time of ending, this schedule to be published in the program. In addition, the Meetings Committee, the Diesel-Engine Activity and the staff of the Society are definitely on record as favoring an open campaign to begin meetings on time and to adhere to a strict schedule which will allow ample time for discussion. No doubt all who attend the Society's meetings will be in accord with this stand. The practice of holding luncheon meetings for chairmen, authors and discussers preceding the Sessions, for last-minute planning of the conduct of each session, was also approved by the Passenger-Car Activity Committee.

J. A. C. Warner and C. B. Veal Appointed

Council Names New Secretary and General Manager and Creates Office of Assistant General Manager

AT A MEETING of the Council in New York City in July, John A. C. Warner was appointed Secretary and General Manager of the Society to succeed the late Coker F. Clarkson, the only previous incumbent of the position, who had held it for the last 20 years. The Council also created the office of Assistant General Manager and appointed C. B. Veal to fill that position. The constantly increasing variety and complexity of the work of the Society have required expansion of the organization and the creation of new offices from time to time. Mr. Warner assumed his new duties the third week in July.

Mr. Warner has returned to the Society from the Studebaker Corp of America, which he has served since November, 1926, as a research engineer in charge of chassis experimental work and as superintendent of the Studebaker Proving Ground at South Bend, Ind. His work with the corporation, in service and production and as a liaison officer with the sales and advertising departments, has involved more than 60,000 miles of travel within the last three years, during which his contact with the field organizations has been extensive. As chassis experimental engineer he has had considerable experience and broad familiarity with experimental work on commercial vehicles as well as passenger types.

J. A. C. WARNER

Born in 1893 at Putnam, Conn., Mr. Warner acquired his public school education in that city. He was graduated from the Worcester Polytechnic Institute in 1917 with the degree of Bachelor of Science in Mechanical Engineering and during the World War completed an officers' course in aircraft and engine design at the Ecole Supérieure d'Aeronautique et de Construction Mécanique in Paris. He has a distinguished record of engineering service in the American Expeditionary Forces in France and Great Britain, and as an engineering officer in charge of numerous projects of the Air Service Technical Section. There he was responsible for the design and construction of Air Service equipment, including an instrument laboratory of which he had charge. He did duty in connection with the inspection and testing of aircraft and aircraft engines for the American Army.

Before entering the Army in 1917, Mr. Warner was engaged at the Bureau of Standards in Washington in the making of physical tests of structural materials used in aircraft. After the war, he became associated with the Borne Scrymser Co. in Boston, spending his time on lubrication and sales problems. Later he returned to the Bureau of Standards as associate physicist. During the following three years he was executive officer and chief of the Aeronautic Instruments Section of the Bureau, and in 1921 was sent as scientific representative of the United States Government to Great Britain, Germany, France, Switzerland and Italy. His familiarity with the foreign languages and his war associations with manufacturing, engineering and Government activities abroad helped to bring success to this important mission.

In 1923 Mr. Warner left the Bureau to become research engineer of the Society, for which he did pioneering work, as Research Manager, in establishing and promoting the early activities of the Research Department. Later he became manager of the Meetings and Sections Department and was responsible for many highly successful National meetings.

Mr. Warner is the author of numerous technical monographs published by the National Advisory Committee for Aeronautics and by trade publications. He has also contributed liberally to the technical and editorial columns of the S.A.E. JOURNAL and TRANSACTIONS of the Society. He has been Chairman of the National Meetings Committee for the last three years, has served on numerous other committees and was nominated for the office of Vice-President in charge of Passenger Car Activities for 1931. He has conducted numerous public relations missions which have involved contacts with Federal, State, municipal and industrial organizations. Through his successful experience and a wide variety of contacts with many activities, Mr. Warner has acquired a very broad acquaintanceship throughout the world with the automotive and allied industries. His selection by the Council as Secretary and General Manager has come as a natural result of this experience and because of his intimate knowledge of the Society's purposes and activities.

Mr. Veal's Broad Experience

Mr. Veal, the newly appointed Assistant General Manager, was born at Greensfork, Ind., in February, 1880, and

C. B. VEAL

was graduated from Purdue University in 1902 with the degree of Bachelor of Science in Mechanical Engineering. For 15 years following his graduation he was retained at Purdue University in various capacities, including instructing in machine and internal-combustion-engine design. He developed the courses in gasoline automobile engines and established the automobile-testing plant there. He was later appointed Professor of Machine Design in charge of the design and testing of motor-cars, engines and parts thereof, as well as the heat-treatment of metals. He remained at the University until 1917. During this period he was also engaged in consulting engineering work and was retained as engineer by builders of passenger-cars, motor-trucks and farm-equipment machinery. He has made many studies in power transmission, production and agricultural engineering fields. Among his clients were the Western Electric Co., McCormick Harvester Co., W. H. McIntyre Co., the Rockwood Mfg. Co., Peru Heating Co., and the Central Station Steam Co. of Detroit. From 1912 to 1917 he was mechanical engineer for the Public Service Commission of Indiana and from 1915 to 1917 was research engineer of the Indiana State Board of Accounts.

At the beginning of the World War he was picked as a special man from a large group of college experts, commissioned as Major of Ordnance to serve abroad, and assigned to the Curtiss Aeroplane & Motor Corp. In association with the eminent aeronautic engineer, Charles M. Manly, he rendered signal service as assistant chief inspection engineer, and at the close of the war became a member of the firm of Manly & Veal, consulting engineers, of New York City. In September, 1926, he was made Research Manager of the Society, a position which he now holds.

Since his election as a member of the Society in February, 1912, Mr. Veal has served as a member or as Chairman of the Constitution, Sections and other administrative Committees. He has been a member of the Grading Committee for several years and Chairman for the last five years. He was a Councilor of the Society for 1927 and 1928. He has served on the Motor-Truck Division of the Standards Committee, was a member of the Operation and Maintenance Committee, representative of the Society on the Engineering Division of the National Research Council, the Committee on Form and Arrangement of Specifications of the American Standards Association, and Division XII on Performance Tests of Rubber Products of Committee D-11 on Rubber Products of the American Society for Testing Materials, and is also Vice-Chairman of A. S. T. M. Technical Committee A on Gasoline. His work in the Research Department of the S.A.E. has been wide and varied, covering cooperative fuel research, headlighting, highway research, riding-qualities and gear research. He has been consecutively active on the committees of the Indiana, Buffalo and Metropolitan Sections and was Treasurer of the Metropolitan Section in 1923 and 1924.

Mr. Veal enjoys a wide acquaintanceship among the nearly 8000 members of the Society and it is believed that his elevation by the Council to the position of Assistant General Manager of the Society will meet with the unanimous approval of the members of the Society.

Chronicle and Comment

General Manager and Assistant Manager Appointed

J. A. C. WARNER was appointed Secretary and General Manager of the Society and C. B. Veal was appointed Assistant General Manager at a meeting of the Council in New York City on July 8.

Members of the Society no doubt will be gratified by the appointment of a man so well qualified and popular as Mr. Warner to succeed our lamented former Secretary and General Manager. Through his former connections with the headquarters staff, as Research Manager and Meetings Manager, and his subsequent activities as Chairman of the Meetings Committee, Mr. Warner is familiar with the varied activities, personnel and routine of the New York City office, and in addition brings again to that office an unusual training and experience in automotive engineering and research work, a broad knowledge of the industry and a personality that is universally liked.

The new General Manager, who assumed his duties on July 19, will have in Mr. Veal an able assistant who, in addition to expert engineering knowledge, possesses sound judgment, intimate knowledge of the detail work of the headquarters staff and great interest in research work.

These two appointments are a recognition by the Council of the growing importance of research in the further progress of the automotive industry and assurance to the membership of the continued able conduct of the work of the Society along the lines laid down and so successfully followed for the last 20 years by Mr. Clarkson. Details of the long experience of Mr. Warner and Mr. Veal in the automotive industry are given in this issue on p. 134.

Vol. XXIV of S.A.E. Transactions

COPIES of Vol. XXIV of the S.A.E. TRANSACTIONS, covering the year 1929, were mailed last month to those members who placed orders for them at the time of paying their Society dues for the current fiscal year. As has been the practice in connection with the last few volumes, a limited number of copies will be available for members who neglected to place an order or those whose copies have failed to reach them. All requests for copies should be made promptly.

Changes, Great and Small

THAT radical changes in automobile design are coming in the near future is a prediction made by William B. Stout at the May 16 meeting of the Detroit Section. He asserted that, with the engineering knowledge the industry already possesses, it is possible to build cars of half the weight of present cars that will have twice their performance in everything except top speed. He indicated that this can be accomplished through body structural design along the lines of airplane practice, and that the roominess of the bodies can be greatly increased by dispensing with running-boards and fenders and extending the body toward the front, laying the engine flat or placing it elsewhere than under a hood at the front.

On the other hand, research and examination into the minutest and most puzzling details of design and operation are going forward. Investigation of the vapor-locking problem, combustion-chamber design and fatigue produced by riding are instances.

The demand for private automobiles seems in little danger of diminishing, as nothing as yet visible on the horizon threatens highway transportation, but the demand for still better cars is persistent.

Those companies that are able to continue to produce vehicles that successively excel their predecessors will reap financial reward. Hence a large field is open for the exercise of both daring initiative in design and in exhaustive research in aerodynamics, chemisty, metallurgy and other branches of science that are involved in the manufacture and operation of motor-vehicles.

Changes in Dues Effective Oct. 1

AMENDMENTS to provisions of the Constitution of the Society relating to annual dues of Members and to Student Enrollment have been approved by letter-ballot of the voting members, as reported in detail in this issue of the S.A.E. JOURNAL, p. 132. The changes in dues go into effect with the beginning of the new fiscal year of the Society on Oct. 1.

Of the annual dues hereafter paid, the Society is to turn over to each Section $5 of the dues paid by each Member, Associate and Affiliate Member Representative residing in the territory of the Section, as provided by changes in the By-Laws.

The effect of the amendments is that hereafter the Society and the Section dues are combined and will be collected by the Society, thus relieving the Sections of collecting Section dues, and that all Members and Affiliate Member Representatives in good standing residing in the territory of a Section automatically become members of the Section upon payment of their annual dues.

Semi-Annual Index with This Issue

THIS issue of THE JOURNAL consists of two sections, Section 1 being the regular monthly issue for August and Section 2 the index to Vol. XXVI, covering the issues from January to June, 1930, inclusive. Any reader who fails to receive a copy of the Semi-Annual Index as a part of this issue of THE JOURNAL is requested to notify the Publication Department at the Society's headquarters in New York City, and one will be mailed to him.

Under the plan for publication of the index for each half-yearly volume of THE JOURNAL, this index should have been Section 2 of the July issue, but the late appearance of the special 25th Anniversary number in June made this impossible. The index to the present volume, XXVII, is expected to be published as Section 2 of the January, 1931, issue.

What Aircraft-Engine Designers Learn from Automobiles

St. Louis Aeronautic Meeting Paper

By ROBERT INSLEY[1]

SPEED, cost, safety, comfort and convenience are the important characteristics that determine the popularity of any means of transportation. The experience of the automobile industry in solving these problems may be of use to the manufacturers of aircraft engines.

The chief asset of the present-day airplane is speed, the other four characteristics listed present problems to be solved. The cost of the airplane engine is a problem of weight, reliability and quantity produced. Assuming that there can be no concessions in reliability, any attempt to reduce the cost materially must consider either an increase in weight or a multiplication of the quantity produced. The quantity depends upon the market and so, for the purpose of this paper, is beyond our control. We are left, therefore, with the two-horned dilemma of high weight or high cost. The safety problem to a large extent is one of mechanical reliability; but it involves one other very serious task, the elimination of fire hazard. Improvement of the comfort factor in air transportation suggests many opportunities to benefit by the experience of the automobile-engine designer in reduction of exhaust and mechanical noise and improvement of vibration characteristics. Similarly, in the matter of convenience, the aircraft-engine designer approaches a problem which is old in the automobile industry, and he faces the necessity of living up to automobile precedent in this regard or establishing new standards of convenience.

Stringent specifications for material and for accuracy of parts was the chief topic of discussion by the engineers who heard the paper. Excessive stringency was laid by some to military influence and to failures which could not be explained by inadequate stress analyses. The Army and Navy were said by others not to have been the authors of stringent specifications, but they may have had inspectors who enforced specifications with little mechanical judgment.

Free service and metallurgical inspection of each bar of steel were cited as large items of cost. Reduction in variety of materials, particularly of special materials, and the adoption of magnetic inspection were said to offer possibilities of reduced cost.

Reduction of noise was said to be impossible with increased propeller speed. Many noises about the airplane itself will be apparent when the engine is made more quiet.

AVIATION has been transformed suddenly from a sideshow to a commercial enterprise. Every aeronautical engineer is now primarily a salesman, and the foremost query in judging the value of an experiment is not Will it fly? but Will it sell? The author hopes, therefore, that he may be forgiven if he dwells upon the commercial rather than upon the purely engineering aspect of this subject.

Five general characteristics—speed, economy, safety, comfort and convenience—determine the popularity of any form of transportation. The automobile, 25 years ago, offered only the first of these as an asset; the other four were liabilities. In speed alone the automobile surpassed its immediate rival, the horse and buggy. The cost of the automobile was nearly five times as great; its safety was openly doubted; its comfort, with single-cylinder engines, canopy tops, noisy exhausts, goggles and dusters and the like, was not even to be compared with the rubber-tired buggy; and it could boast little of convenience at that time. Yet, on the basis of speed alone, it was sold to an eager public in rapidly increasing numbers.

Today the airplane finds itself in much the same position with respect to the automobile. Its chief asset is speed; it is vastly more expensive in first cost and in maintenance; its safety is openly questioned; its comfort and convenience, to any but the dyed-in-the-wool flier, leave much to be desired. The parallel is perfect up to this point. The automobile industry worked out its problems and converted its liabilities, one by one, into assets.

Those of us who are interested in the regularity of aeronautical meals must set about vigorously to see what can be done about the cost, safety, comfort and convenience of the airplane. The problem belongs to every man in the business, from advertising manager up or down to the men at the lathe and drill-press; but a large part of the responsibility lies at the door of the engine manufacturer, and it is to his part of the task that this paper is directed. With the successful evolu-

[1] M.S.A.E.—Vice-president, Continental Aircraft Engine Co., Detroit.

tion of the automobile in mind, it is not unreasonable to hope that some clues to improvement in the marketability of aircraft engines may be discovered by a cautious examination of the relation between the two types of powerplant.

Cost Puts Builders on the Defensive

Automobile and aircraft engines are largely identical in general construction and in operating characteristics. Both acknowledge the same ancestry and have followed a similar process of evolution in their development; but they serve fields of usefulness differing in several inherent characteristics, and they now cater to markets of vastly different proportions. These two conditions are responsible for the dissimilarity between engines of the two types and must be kept carefully in mind in their comparison.

Difference in cost is the most disturbing difference between automobile and aircraft engines. A general complaint about the excessive cost of flying has developed since recent historic flights have impressed upon the public mind the commercial possibilities of air transportation, and it has been generally rumored about that the cost of the engine is chiefly responsible. It has become common knowledge that the automobile engine represents approximately 10 per cent of the cost of the car, whereas the aeronautic engine cost is more nearly 40 per cent of the cost of the completed airplane.

Many individuals—including, unfortunately, a number of airplane manufacturers—have become disturbed by the discovery that a high-grade automobile-engine of 100 hp. costs the car manufacturer something less than $300, while the lowest-priced airplane-engine of the same power cannot be purchased for much less than $1,200. The condition seems to demand either an explanation or a violent price revision on the part of the engine manufacturers. While making such an explanation, I shall attempt to suggest what opportunities there may be for price reductions.

Weight, reliability and quantity are three words that give the reason for the comparatively high cost of aircraft engines. The aircraft engine propels a vehicle which is supported in the air by virtue of that propulsion. The airplane engine must essentially be as light and powerful as is possible, consistently with safety. Obviously, no such limitation restricts the automobile-engine designer. The aircraft-engine designer therefore employs every device at his command to improve power and reduce weight. He hesitates at no increase in cost, within reason, that will assure a corresponding increase in power or reduction of weight. He employs the lightest materials which will serve the purpose of the part in question. Aluminum and magnesium alloys for structural parts cost more than cast iron; fancy steels for highly stressed working parts are purchased only at a premium and require the most minute visual and metallurgical inspection before approval. Special equipment such as superchargers, required in some cases to improve performance under special conditions, adds to the complication and the cost.

Why the Designing Costs More

The design work itself is more expensive than that of the automobile engine, not only because of the scarcity of competent aircraft-engine designers and the obvious result of such scarcity, but because every aircraft engine is a new development problem and requires months of careful study. The designer computes his stresses to take every advantage of the material used.

Fully 75 per cent of the new automobile engines are very nearly identical with some previous model and therefore involve little original study; but aircraft engines are not yet so standardized, and each new model is a new problem. Because of these facts, the development tests to assure a satisfactory product are also more elaborate. The designer, to secure the minimum of weight with the maximum of power, takes liberties with stresses and bearing loads which require very extensive testing before they can be released to the public.

In actual construction cost also the customer pays for light weight and quality. To secure the lightest weight without sacrifice of reliability, highly stressed parts are completely machined, thus removing excess material and surface imperfections. The principal difference in actual labor required lies in the removal of excess material. There is a generally erroneous impression that the working parts of aircraft engines are held to closer tolerances than those of automobile engines. That is true only to the extent that the tolerances control the weight of parts.

Weight Removal Is Major Cost

Careful analysis of a high-grade automobile-engine design will disclose the fact that cylinder sizes, piston fits, bearing clearances, gear backlash and similar working fits are held as closely or even more closely in some instances than the corresponding fits of the aircraft engine. It will be found generally true that such parts as gears, pistons and bearings, which are likely to produce disturbing noises, are held to closer limits in the automobile engine than are necessary for satisfactory service in the aircraft engine, where silence is not at present an important sales factor. But the removal of excess material to reduce weight on such parts as connecting-rods, crankshafts, cylinders, and accessory-drive gears and shafts, where little attempt is made to reduce weights in the automobile engine, not only increases the actual labor cost of the aircraft engine but adds immensely to the cost of tooling required, makes dimensional and metallurgical inspection much more elaborate operations, demands a more exacting system of acceptance testing and throws a vastly greater service responsibility on the manufacturer of the aircraft engine.

It is clear that, so long as the aircraft industry demands an engine of the lowest possible weight, and having dependability and durability equal to or better than that of the best automobile engine, the cost of design and development, of tool equipment and manufacturing labor, of materials and inspection, of tests and servicing must be distinctly greater than the corresponding costs of the automobile engine. But there is more to the story than that. Automobile engines are manufactured in comparison with aircraft engines at the rate of nearly 1000 to 1.

Quantity Introduces Cost Differential

Everything else being equal, the cost of manufacture of the airplane engine on the score of production quantity alone would be immensely greater than that of the automobile engine. The automobile engine is designed with immense production in mind. The designer is free to take advantage of elaborate special machinery, multiple-spindle boring-machines, enormous milling

equipment, marvelous camshaft and crankshaft-turning, milling and grinding machines—production machinery which cuts labor hours to minutes. He designs instinctively to such machines. Aircraft-engine production justifies only such special machinery as is essential to the accuracy of the job. The operations in the manufacture of aircraft-engine parts must be adapted, so far as possible, to general-purpose machines.

On the one hand, the aircraft-engine designer has greater latitude in design, because he is not required to work so closely to production requirements; but, on the other hand, it frequently occurs that he must modify the design seriously to adapt it to existing machines. The net result is that, though the total tooling cost of the automobile engine may amount to as much as ten times that of the airplane engine, the ratio of production cuts the overhead represented by tools for the automobile engine to a small fraction of that for the aircraft engine. Correspondingly, the overhead due to such items as supervision and fixed charges is lower in automobile-engine production. An estimate that aircraft-engine overhead is four times that of the automobile engine may be regarded as conservative.

Weight, reliability, quantity and cost are balancing factors in aircraft-engine production. It is unreasonable to expect any great improvement in any of these factors without some compensating sacrifice in the others. One of them is beyond the direct control of the designer; the quantity of engines built can be no greater than the capacity of the market and, for the purposes of this discussion, that factor can be eliminated. However, several concessions in the other three factors are possible which may reduce cost without seriously affecting weight or reliability.

Can Extreme Lightness Be Sacrificed?

Engine manufacturers often have reason to suspect that engine weight is a matter about which the aircraft operator likes to be deceived. Great stress is laid on the minimum weight of the bare engine, and that engine manufacturer who can conscientiously quote the lowest bare-engine-weight figures gets off with a great advantage over his competitors. Once the engine is sold, the operator seems perfectly willing to hang upon it untold quantities of exhaust piping, silencers, hot-spots, air heaters, generators, starters, cockpit and cabin heaters, air cleaners, oil heaters and coolers, and other accessories to obtain satisfactory performance of the engine. It is natural to wonder, if the purchaser is so vitally interested in cost and apparently so little concerned about the weight of accessory gadgets, why he should object to adding a few pounds of internal weight if by so doing we can materially reduce the cost without affecting reliability.

Can we not swallow our pride and tear a leaf from the automobile manufacturer's book? Unmachined connecting-rods are heavier than machined rods and are commonly regarded by the aeronautical mind as untrustworthy, but try to recall when you last heard of an automobile connecting-rod failure. Plain forged heads having what is known as restruck finish stand up very satisfactorily in automobile service, but are regarded as too heavy and unsafe for aircraft engines. We spend many dollars per engine in reaming and grinding the inside of piston-pins, link pins and shafts, whereas we might copy automobile parts if we were content to add a few ounces. Valve-gear weight in the aircraft engine is considered very important, but few people wonder why we go to such pains to machine valves all over and leave the rocker arms as forged. Cast-iron cylinder-barrels, integrally forged and unmachined crankshaft counterweights, and a dozen other compromises on weight that would not reduce reliability might be considered. Such practices would be a severe blow to the engine designers' creative pride and would arouse little enthusiasm in aeronautical circles; but, if the crying need of air transportation really is cheaper engines, one solution of the problem lies in that direction. I do not wish, however, to go on record as heartily endorsing such expedients.

No Compromise with Safety

With our winter crop of aircraft fatalities coming off the press, the factor of safety in aircraft-engine design probably is even more important to saleability than is the cost factor. There is no direct comparison with automobile practice in the matter of safety. The design of the engine has slight effect upon the safety of an automobile; the engine designer can do little to either increase or lessen the safety of the motor-car. But the safety of the airplane depends directly upon its powerplant, failure of which frequently means disaster, and the airplane-engine designer therefore has a responsibility which has no parallel in automobile-engine design. The fact that only about 15 per cent of the aircraft fatalities are directly chargeable to powerplant failures is a consolation but not a sinecure. There can be no compromise with safety, and it behooves aircraft-engine designers to bear this fact in mind constantly.

Mechanical failures form an almost insignificant percentage of the causes of fatal accidents, and there seems to be no decided trend in such mechanical failures as occur. The valve gear seems to be a rather troublesome feature in radial engines, and we may look forward confidently to some drastic improvement in that regard. Failure of fuel and lubrication systems divide the honors about equally. Unfortunately, the responsibility for the two systems seems also to be divided about equally between the manufacturers of the airplane and the engine, so we shall probably have to look to the Army and Navy for salvation.

The most pressing of powerplant-safety problems and that which affects public enthusiasm most seriously is the problem of fire hazard. Fires in the air from stray sparks and leaking fuel lines and fires after relatively minor crashes from which there is reason to believe that the passengers might otherwise have escaped will strike terror to the heart of the most enthusiastic prospect. Some sort of low-grade fuel, whether used in Diesel or in explosion engines, seems to be the most likely solution of this problem at the moment. Upon the engine manufacturer rests an obligation to eliminate this hazard as early as possible.

Reducing Noise and Vibration

Comfort affords a particularly interesting comparison between aircraft and automobile-engine design. Here the automobile-engine designer for years has battled a problem which is now just confronting the airplane-engine designer. Noise, vibration, smoke and oil have been all but forgotten in the thrill of flying, but aircraft must be comfortable before they can be

popular as common carriers. This is a problem which primarily affects the engine designer, for most of the objectionable features of the airplane originate in the powerplant. The annoyance of exhaust noise will be eliminated by the use of mufflers, many of which are reported available but few are actually in use.

Reduction of exhaust noise will uncover mechanical noises, of which there are plenty. The aircraft-engine designer in this instance may profit directly by automobile experience. Before many months, he must concern himself with piston clearances, valve-gear clatter, quiet cam contours, combustion-chambers, gear backlash, resonant diaphragms—all problems which the automobile-engine designer has solved so successfully to produce the modern almost silent automobile engine. Fortunately, that experience is available to build upon, and the problems should be relatively simple.

Relatively light airplane and engine structures make troubles of the airplane-engine designer with regard to vibration somewhat more difficult than those of the automobile-engine designer. The vibration problems of the automobile engine have been solved to a large extent by the addition of weight at strategic points in the engine and the supporting structure. This course is open to the airplane-engine designer only to a limited extent. To satisfy the desires of the air passenger, we may be driven to drastic alterations in the general type of engine to obtain the best inherent balance. The problem is involved directly with the noise nuisance, and only when noise troubles have been eliminated will it be possible to discover the objectionable influence of vibration. Motor-car manufacturers almost universally have adopted rubber mountings to absorb engine vibration. The aircraft industry already has profited to some extent by this experience and undoubtedly will find it profitable to carry the practice further.

Convenience Should Be Improved

Aircraft-engine designers have much yet to learn from the automobile fraternity in the matter of convenience. The automobile owner, accustomed to relatively care-free transportation, will not long be content with airplane engines which are difficult to start in cold weather, which require fresh, warm oil every winter day, and demand minute inspection and maintenance service every few hours. The automobile driver will protest violently before he will submit to ownership of an aircraft engine which cannot be buttoned up in the cowling and operated for months with only routine station servicing. Whether or not this is reasonable, the automobile driver has been trained to expect such service, and the aircraft-engine manufacturer must reconcile himself either to the task of providing such an engine or to that of re-educating the user.

THE DISCUSSION

ARTHUR NUTT[2]:—How many of you have tried talking to another occupant of your car while driving it at say 80 per cent of its maximum speed? The automobile man has certainly improved his car so that it is very pleasant to ride in at 25 or 30 m.p.h., but he has the same noise problem that we have at high speeds.

After all these engine noises are eliminated from the engines of the present cabin monoplanes, a surprising amount of noise will remain, from such sources as vibration of the cabin walls, window rattling, whistling of struts and interference between struts and wires. Certainly the airplane designers then will have worries of which they have never thought before, because these noises have been drowned in noises of the exhaust and the propeller.

Propeller noise is greatly reduced by the use of slow-speed blades; but I have recently observed a tendency, particularly in the smaller engines, to use a high speed as the basis of rating. Some of them are rated at 2300 to 2400 r.p.m., with direct-drive propellers. This imposes an impossible condition as to noise. Discussion of the relative merits of the low-speed direct-drive engine and higher-speed geared engine would be in order.

J. H. GEISSE[3]:—In my opinion, weight requirements are more important than quantity in the cost of airplane engines. It has been said that a certain automobile engine costs the manufacturer only about $35; the forgings alone for an aviation engine cost more than that, because all of them are made of special steel, and the grain flow must be controlled to keep down weight.

Money spent in reducing the weight of an airplane engine results in reducing the cost of the operation of the airplane per pound of payload, but airplane manufacturers seem to be less interested in that than in the original cost of the engine. If the industry is to survive, the cost of transportation must be reduced. The original cost of an airplane is not so important as keeping it sold.

ROBERT INSLEY:—I did not mean to suggest that the cost of an aircraft engine could be reduced to that of the automobile engine by quantity production; but probably the material and labor in a $2,000 aircraft engine, bought in reasonable quantities, would not cost more than $600 or $700, the rest of the cost being overhead. If the overhead can be reduced to 100 per cent of the labor cost, the engine can be sold for $1,000, still at a little profit.

Unwise Specifications Add to Cost

G. C. BROWN[4]:—The high cost of aircraft engines is pieces and cores in one section and 36 loose pieces and dimensional, of the materials required. I have seen crankcase castings, sound in every respect, rejected on account of a variation of 1/64 in. in the wall thickness, which variation would not affect the serviceability of the engine.

Recently we had crankcase patterns with 47 loose pieces and cores in one section and 36 loose pitces and 18 cores in the other section. No automobile manufacturer would think of sending such pattern equipment to a foundry. Redesigning the latter section eliminated the 36 loose pieces and left only the 18 small cores for the valve gear. This change made possible a tremendous increase in the number of castings produced per day, as well as reducing the rejections. A marked reduction in cost was also a desirable result.

Airplane-engine crankcases cannot be made from the

[2] M.S.A.E.—Chief engineer, motor division, Curtiss Aeroplane & Motor Co., Inc., Buffalo.

[3] M.S.A.E.—Vice-president in charge of engineering, Comet Engine Corp., Detroit.

[4] M.S.A.E.—Sales engineer, aircraft division, Bohn Aluminum & Brass Corp., Detroit.

same materials that are used for motor-car engines on account of the weight limitations, with the resultant higher stressing of the metal. If the time ever arrives when the same materials can be used in aircraft engines that are used in automobile engines, the cost of the aircraft engine will be materially reduced.

The inspection of the various items entering into the construction of airplane engines is very rigid. We make bronze valve-seats having a Brinell hardness between 200 and 220. To make sure that the entire lot will come within the specifications requires the services of a trained metallurgist and the testing of a large percentage of the castings.

The duty required of automobile and aircraft engines is so different that there appears now to be no possibility that aircraft engines will be produced as cheaply as automobile engines.

E. D. HERRICK[5]:—Volume production eventually will reduce the cost of the materials and we are all seeking a way to reduce the specification requirements. The extremely rigid specifications for materials have been very beneficial, and the benefit will be reflected into the automobile industry. Materials are being improved all the while, and the prices can be reduced through quantity production.

MR. NUTT:—Reduction in the cost of material will come not so much through quantity production as through a reduction in the number and cost of the materials specified. More than 40 different kinds of material now are specified for an aircraft engine. We must try to reduce the number, to eliminate the most expensive materials, and to adopt materials that are used in the manufacture of other commodities. This will increase the quantity of material on various specifications and give us the quantity benefit of the outside production.

Military Influence Can Be Hindrance

P. B. TAYLOR[6]:—The industry is suffering to a large extent from the military influence on engine construction. It is impossible to make the number of core openings in a crankcase smaller when provision must be made for two gun synchronizers, two tachometer drives, fuel-pump, oil-pump, starter, generator, possibly two magnetos and perhaps a few other attachments that require drives of appropriate ratios.

Military insistence on perfect castings, held to tolerances of plus or minus 1/32 in., has influenced the inspection of commercial engines by a man who is accustomed to receiving a severe calling down for every deviation from the blueprint. Undoubtedly the inspection requirements can be reduced in many respects for commercial service, but they should be made more rigid in other respects. We are undoubtedly too severe in regard to sections of castings, but no defects should be passed which can possibly cause fatigue failures. The addition of two or three pounds of aluminum might reduce the cost of the engine considerably.

MR. GEISSE:—Special runs at the mill are a minor item. The big items in increasing the cost are things like the special microscopic inspections that are required at each end of every billet, which involve laboratory work. Devising some means by which that inspection can be made on a production basis would reduce the cost.

Increasing the quantity may reduce the overhead as to sales and office expenses; but, unless service is to be given on the same basis as for automobiles, the service overhead will not be reduced. Now, when a part breaks, the engine maker is obliged to replace both the part that may have been defective and other parts that were broken, without cost to the operator for either material or labor.

W. F. DAVIS[7]:—Readjustment is needed both in the material list and in the inspection requirements for aircraft engines. We have reduced a material list of 40 or 50 different steels and aluminum alloys to three or four steel specifications and two or three specifications for aluminum alloys for the whole engine. Another cause of the high cost of materials is the fear on the part of the vendor that rejections will be excessive. A widespread mental readjustment is required.

The Army and Navy specifications, which govern almost all aircraft-engine activities, were absolutely necessary at the time they were first imposed. Mills are now turning out steels of much better quality and greater consistency, which need less inspection. Our inspection methods need to be improved and simplified to make the desired economy possible.

J. H. GEISSE

Material Is Convenient Alibi

ROLAND CHILTON[8]: — Unexplained breakages were very frequent when we began to build light engines. We thought that extra quality in the materials was the magic that would cure our troubles. As we developed in the art, we found perfectly logical reasons for part failures in some action of the engine that we had not previously fully understood. Our calculations had failed to include all the stresses to which a part was subject. If the part failed, we were too proud to admit that our calculations were wrong, and we blamed the material.

If it were possible to analyze 100 typical failures that were considered, five or ten years ago, as being due to the materials, we would find that not more than two or three of them really were the fault of the material, they were due to our lack of understanding of the stresses to which the parts were subjected. Specifications having unnecessary items have gone all through the industry; for instance, inspection for surface finish has been insisted upon for parts on which it is of no importance. One reason is for the psychological effect, so that an inspector will not get into the habit of passing defects on unimportant parts and then pass similar defects on important items.

The automobile manufacturer has an immense advantage in the experience he has gained from building enormous quantities of engines; he has found out from bitter experience which parts need the superior mate-

[5] M.S.A.E.—Chief engineer, Lycoming Mfg. Co., Williamsport, Pa.

[6] M.S.A.E.—Acting chief engineer, Wright Aeronautical Corp., Paterson, N. J.

[7] M.S.A.E.—Chief engineer, Fairchild Engine Corp., Farmingdale, Long Island.

[8] M.S.A.E.—Consulting engineer, Wright Aeronautical Corp., Paterson, N. J.

rials and what failures can be prevented by more intelligent design. As a result, very few parts of an automobile are subject to material failures. The same will apply to the aircraft industry when it has a corresponding volume of experience behind it.

Danger in Double Standards of Quality

G. D. WELTY[9]:—The remarks of Mr. Chilton have been very much to the point. I can recall numerous cases of service failure of aircraft-engine parts in which the difficulty has been attributed to material when it was in reality faulty design or improper machining practice, as determined by a more careful analysis or by subsequent experience. The true cause of failure frequently is obscure and "faulty stock" or "improper heat-treatment" are too often resorted to as sufficient explanation, with the result that material specifications are tightened up to no purpose, thereby increasing costs to both engine manufacturer and suppliers of rough or semi-finished parts.

ROLAND CHILTON

I do not wish to be understood as making a plea for greater looseness in material specifications. These should be held as rigid as necessary in order to maintain a uniformly high-quality product. What I am opposed to are specifications which do not improve the quality but which do tend to increase manufacturing costs.

A more careful analysis of failures, together with constantly increasing experience and knowledge of material properties, will go far toward increasing engine reliability and decreasing costs.

MR. GEISSE:—I wonder if the reduction in the number of failures attributable to material is not due largely to the fact that materials now are much better and much more rigidly inspected than they formerly were.

MR. CHILTON:—I still feel that such improvement is only part of the story. The temptation is great for an engine designer to place the responsibility for a failure upon the material instead of upon an error of his own. If the crankshaft is operated at its torsional period, no improvement in material will prevent the destruction of the shaft or of the gears that it drives.

One of the greatest recent advances is the magnetic bar-stock analyzer, which I have described[10]. If we had some such means for determining that the forgings for highly stressed parts are free from serious defects, we would be still further along. Present material specifications do not provide for any such inspection. We may examine a sample from one corner of the forging and overlook a serious defect only a small fraction of an inch away from it. If our material specifications were concentrated on the elimination of defects that start fatigue failures, many other tests could well be discarded. The characteristics upon which we now insist in our material specifications are not always the most vital ones in their relation to the performance of the part.

MR. INSLEY:—I still feel that there is plenty of room for improvement in the cost of airplane-engine parts, and I should like to suggest a complete discussion of airplane-engine costs at a future meeting of the Society by someone who is well versed in the subject.

Specifications Not Written by Services

CHAIRMAN B. G. LEIGHTON[11]:—Although I am not now a Service man, I have been connected with the Service for a long time and I have seen both sides of this question. A great tendency exists to blame the Army and Navy for any trouble, and not all of it is their fault. During the 3½ years while I was concerned with purchasing engines in the Bureau of Aeronautics of the Navy Department, our constant statement to manufacturers and designers of aircraft was: "Design us an engine to meet a certain requirement and submit the design to us. It may be made of wood, sawdust, brass, copper or steel." In most cases such engines were developed with funds made available by the Department, and no Naval inspectors participated in the initial designs. No restrictions were placed on the manufacturers as to materials and designs they should use; all we asked for was the result.

When an engine had been developed and accepted after performance tests, however, we did insist that the manufacturers should submit detailed specifications as to material, heat-treatment and dimensions showing how they got their results with the experimental models, and when buying additional engines we always insisted on rigid adherence to those specifications, which were made by the manufacturer himself, to assure that results should continue. I am fully aware that some inspectors interpret these specifications too rigidly, insist on the letter without reference to engineering sense, and sometimes hold up production and cause unnecessary costs, but as a representative today of a commercial manufacturer, I am convinced that this situation is not nearly so bad as it is sometimes represented to be.

At any rate we owe much to the Army and Navy for initial research and development and for the money that was contributed to the early development of the aircraft industry, and that is still largely supporting advanced developments today.

MR. NUTT:—It might be of interest to know that our organization has never built an aircraft engine for the Army or the Navy to Army or Navy specifications; they have all been built to Curtiss specifications. The whole trouble is in the interpretation of those specifications by persons who have not had sufficient experience or qualifications.

[9] M.S.A.E.—Engineer, Aluminum Company of America, Cleveland.

[10] See S.A.E. JOURNAL, June, 1930, p. 771.

[11] M.S.A.E.—Director, sales and service, Wright Aeronautical Corp., Paterson, N. J.

In-Line Liquid-Cooled versus Air-Cooled Engines

Metropolitan Aeronautic Meeting Paper

By GEORGE J. MEAD[1]

THE PAPER deals mainly with conditions more or less generally applicable to commercial aviation. After reviewing the general development of aviation powerplants and analyzing the reasons behind the changes in engine types as a guide to what may develop for the future, the author goes into some detail as to the powerplant requirements of commercial aviation of today, mentioning dependability and minimum weight per horsepower as the most important.

Speed is one of the principal advantages of aviation, and he says also that we can expect strenuous competition between the various transcontinental lines in this regard, which will not only force close scrutiny of the way in which the available power is used, but will direct attention to the development of larger and larger power-units.

Among other commercial requirements, it is stated that passenger comfort needs to be improved, which will necessitate quieting the propellers by reducing their speed with gearing. The engine exhaust will need to be muffled and conducted well aft of the passengers, and engine or propeller vibration will not be tolerated. In the author's opinion, engine units of greater power will have a larger number of cylinders than at present so that vibration will be reduced to the minimum. Costs will always be important as regards powerplant fuels, maintenance and depreciation.

Assuming that larger power-units are apt to be favored, the author states that experience seems to show that 600 hp. is the maximum power which can be applied efficiently to one crankpin. He doubts that cylinders radially disposed in more than two planes can be cooled satisfactorily, and says that it seems that we cannot predict with any certainty whether the liquid-cooled in-line engine or the air-cooled type will be favored for the larger power-units. Whichever type becomes standard in the larger powers, the weight of the powerplant per horsepower will be comparable with that of the air-cooled radial unless a gain in performance is secured because of reduced engine-drag sufficient to compensate for its additional weight. In his opinion, the standardization of new types is bound to take considerable time both for thorough development and for service testing.

Various points brought out in the discussion included statements that greater cooperation between aircraft and engine designers is needed and that present conditions justify placing of great emphasis on the subject of improved appearance and greater attractiveness of airplanes so that they may make a stronger appeal to the flying public.

GEORGE J. MEAD

NO COMPARISON of engine types can well be made without first considering the history of each and then the manner in which it fulfills the requirements of commercial aviation. Therefore, without especially touching on the military aspect of this question, let us review the general development of aviation powerplants and analyze the reasons behind the changes in engine types as a guide to what may happen in the future.

The initial effort in powerplant design was directed toward an engine light enough to permit flight. Steam was tried, but was found wanting. Finally, the then new and more or less untried internal-combustion engine made the first flights possible. Since then it has been a continual battle to reduce powerplant weight. Weight saving gave us the first power flight, and subsequent savings have provided for the payload without which we could have no commercial aviation.

The engine used by the Wright brothers had four vertical water-cooled cylinders and developed 30 hp. Its weight was approximately 6 lb. per hp. Following this pioneer effort, all sorts of cylinder arrangement were tried, with both water and air cooling. It was only natural that the current types of automobile engine should be given serious consideration. It was found, however, that the four-cylinder engine then in vogue was too heavy and rough. It must be remembered that in these early days the technique of handling aluminum

[1] M.S.A.E.—Vice-president, Pratt & Whitney Aircraft Co., Hartford, Conn.

had not been developed. Consequently, cylinders were made of cast iron and this resulted in a heavy engine. By 1912, however, the air-cooled rotary-engine was more or less the accepted standard. These engines weighed between 3 and 4 lb. per hp. Air cooling permitted the use of steel cylinders with integral fins, and the radial form made it possible to construct the crankcase of steel also. These engines remain a masterpiece of shopwork, since practically every piece was machined from the solid metal. At this time the speed of the airplane was comparatively low, so that the rotary motion was used in an effort to cool the cylinders. The power, however, was limited because of centrifugal force so that, when more than 200 hp. was required, it was necessary, therefore, to resort to a different type of engine.

In 1914 the German Mercedes Company developed welded-steel cylinders, under the direction of Mr. Friz. This innovation, with the use of aluminum for crankcases, brought the weight of in-line six-cylinder engines down to approximately 3½ lb. per hp. This opened the way to greater powers with the in-line and the V types of engine. Soon after this Mr. Berkee, the noted Swiss engineer, developed the Hispano Suiza engine. He used a built-up type of cylinder construction in which the cylinders were closed-end steel-sleeves screwed into aluminum blocks which formed the water-jackets and ports. The original engines of this design developed 120 hp. and, before the end of the war, engines of 300 hp. were being built with a weight of less than 2.25 lb. per hp.

In 1917, the Liberty engine was developed. It used built-up steel cylinders and aluminum crankcases which brought the weight of the 400-hp. engine down to approximately 2 lb. per hp. The necessity for greater power had forced the reduction of weights of types hitherto undesirable for aviation so that, by 1919, the in-line and the V water-cooled types were practically standard.

Reduction of Powerplant Weight

Beginning in 1920, however, commerial operation showed the need for greater reduction in powerplant weight as well as increased dependability. The British, during the war, started the development of fixed-radial air-cooled engines. They saw that this type gave promise of much better plane performance because of the reduced powerplant weight. This meant good high-speed performance, with exceptional climb. Their work, together with the demands of commercial aviation, stimulated the radial air-cooled development in this Country so that, by 1922, the air-cooled radial was coming into its own. For the last 4 or 5 years this type of engine has been practically the standard powerplant for our commercial flying. Powerplants of this type having approximately 600 hp. are now being built; their weight is approximately 1½ lb. per hp. It might truthfully be said that the use of aluminum alloys has made possible the low weights per horsepower of the radial engine.

It will be seen from this brief survey of powerplant history that the demand has always been for greater power with less powerplant weight. In practically no instance has a new type been accepted unless the weight was at least equivalent to that of its predecessor. Hand in hand with the reduction in weight has come greater dependability.

Present Powerplant Requirements

It is next important to consider the powerplant requirements of commercial aviation as we know them today. Besides dependability, the most important requirement is minimum weight per horsepower. It used to be said that low weight was not essential for commercial powerplants. The fallacy of this reasoning has been proved by the way aviation has embraced each lighter type of powerplant as it was developed. Obviously, the powerplant's dependability must not be sacrificed to reduce the weight. Fortunately, it has been found that mere weight was not synonymous with dependability. In view of this, it seems reasonable to believe that commercial aviation will not support an engine type which reduces the possible payload that may be carried with a particular size of powerplant.

For example, the Diesel engine may add ¾ lb. per hp. In a 500-hp. unit, which is standard for our transcontinental lines, this would mean a reduction in payload of 375 lb., against which, however, must be credited the saving in fuel which might amount to 125 lb. for a 3-hr. flight, which is the longest now made by commercial-aviation planes. This shows that the payload will be reduced 250 lb., which very seriously cuts into the revenue of the operator, since the cost of the fuel saved amounts to approximately $15, while the revenue for the payload that might have been carried during the 3 hr. could have been as much as $130.

Speed Considerations Outlined

One of the principal advantages of aviation is speed. Consequently, we can expect strenuous competition between the various transcontinental lines in this regard. This will not only force close scrutiny of the way in which the available power is used, but the development of larger and larger power-units. From a survey recently made, it appears that 30 per cent of the available power is now used to overcome unnecessary resistance which could be eliminated, thus making this power available for the propulsion of the plane. It is well to emphasize this point, since our present 600-hp. engines could give a properly designed plane practically the speed that could be obtained with our present plane and 800 hp. Very roughly, this would account for an improvement of 20 m.p.h.

In this Country several planes are already in operation, equipped with as much as 2000 hp. The Do-X in Germany used 6000 hp. Obviously, it would be much simpler if we had larger power-units rather than having to install and maintain 12 separate powerplants as is the case with the Do-X. It must be remembered in this regard, however, that the powerplants should be of such size and number for each plane that there is ample reserve power so that the plane can be operated on 75 per cent of its total power when cruising, or in the event of failure of any of its engines.

Passenger Comfort and Operation Cost

Passenger comfort is another matter which will come in for a great amount of improvement in the near future. This will necessitate quieting the propellers, which can be done by reducing their speed with gearing. The engine exhaust must be muffled and conducted well aft of the passengers. Engine or propeller vibration will not be tolerated. With greater engine-powers, it is natural to expect that each unit will have a larger number of cylinders than at present, not only to obtain

the greater power, but to reduce vibration to the minimum. It is well for anyone who doubts the necessity for these improvements to consider the history of rail transportation, and to note carefully the improvement that has been made in answer to the passengers' demand for greater and greater comfort.

The cost of operation will always be an important factor. In analyzing costs, the important items to which the powerplant contributes are depreciation, maintenance and fuel. Fuel costs average only 10 per cent of the total cost per mile, so that an improvement in this direction, although helpful, will not make any startling change in the cost of operation. This figure may be most improved at the moment by building longer life into the equipment, thereby cutting down the depreciation, and by reducing the maintenance required.

Tendency toward Larger Powerplants

As a result of the foregoing analysis of the commercial requirements, it seems that larger powerplants are sure to be required in the near future; moreover, that powerplant weights are extremely important and that whatever type of engine is chosen will have to approximate the standard weight per horsepower established by the air-cooled radial. What type, then, is apt to be favored for the larger power-units?

From experience to date it seems that 600 hp. is the maximum which can be applied efficiently to one crankpin. It seems to follow that larger units could be developed having several crankpins. Air-cooling, however, seems ideal only for cylinders disposed in a single radial plane and possibly quite satisfactory for cylinders disposed in two planes, one behind the other.

It is doubtful whether cylinders radially disposed in more than two planes can be satisfactorily cooled. There are numerous examples of in-line and V-type air-cooled engines, and it is possible to build such types in fairly large sizes. Unfortunately, however, the weight per horsepower of these cylinder arrangements is high when air-cooling is used. In this Country, we have found it necessary to use single inlet and exhaust valves to get the best results from air-cooled cylinders, since the four-valve heads could not be satisfactorily cooled. Therefore, the speed of revolution is restricted with large-bore cylinders, because of the reciprocating weight of the valves and gear. This limitation can be overcome by the use of more cylinders having smaller bores and therefore lighter valves.

With our present method of air cooling it certainly seems possible to build engines of 800 hp. Today, within its power limitations, the fixed-radial air-cooled engine gives the lightest powerplant for commercial aviation; at the same time it is dependable. The future development of direct cooling may show the way to reduction of engine weight with other cylinder arrangements than the radial, so that it is impossible to predict the maximum power of the air-cooled engine that may be available in the next 5 to 10 years.

Types of Cooling Outlined

With comparable design and equal durability, the weight per horsepower of the radial air-cooled engine and the V-type water-cooled engine without its cooling system are practically the same. The water-cooled engine must pick up an additional 0.6 lb. per hp. for its water and radiator, which is a serious handicap. Already, much work has been done in regard to ascertaining the possibilities of reducing the weight of the cooling system.

The latest development along this line has been the use of ethylene glycol instead of water. Such a system operates at 250 to 300 deg. fahr. instead of at 160 to 180 deg. fahr. Consequently, this reduces the size of the radiator required, which, in this case, averages approximately 0.35 lb. per hp., including the liquid in the system. It has been found, however, that it is no longer possible to cool the oil by means of the circulating liquid on account of its high temperature in the ethylene-glycol-cooled engine. Oil radiators must, therefore, be added which weigh approximately 0.05 lb. per hp. so that the total weight of the cooling system may be 0.40 lb. per hp., giving a theoretical saving of about 30 per cent. It is doubtful, however, whether equal durability will be obtained unless certain parts are strengthened to withstand the higher operating-temperatures, which, in turn, may cancel some of the saving in weight in the cooling system.

Water or liquid-cooling has the advantage in that the cylinders may be equally cooled almost without regard to their arrangement. Moreover, a number of small valves per cylinder can be used satisfactorily, which permits higher crankshaft-speeds than are possible with the air-cooled cylinder of the same displacement because of its heavier single valves. Since reduction gears will almost certainly need to be used to reduce the noise of the propeller, higher engine-speeds can be taken advantage of, provided that the present standard of dependability can be maintained. Unfortunately, it has been found that, although higher engine-speeds show the way to greater power for a given displacement, the weight per horsepower is not materially altered. This is not surprising when one considers the stresses involved, and we find ourselves again confronted with the old maxim that we never get something for nothing. This makes it difficult to design a water-cooled or a liquid-cooled powerplant which will have a weight equivalent to that of the present air-cooled type, since not only must the cooling-system weight be reduced, but the engine weight as well.

Reduction of Powerplant Drag

Ways may be found of reducing powerplant drag to such an extent that, even though the type of engine involved weighs more per horsepower than the present standard, it makes available a greater percentage of its power for the propulsion of the plane. At present, the drag of a radial air-cooled engine with suitable cowling is at least no greater than that of a V-type water-cooled engine of equal horsepower with a tunnel radiator. Wing radiators no doubt will be developed which will be durable. These will in part compensate for the weight of the water-cooled engine.

Fuel saving is important from a weight-saving standpoint as well as from that of expense, but this again must be considered in the light of the total powerplant weight. From the recent fuel developments, it seems that much better fuel-consumptions will soon be available. It seems rather unfair to compare the Diesel cycle operating on compression ratios of 14:1 with our present engines running on the Otto cycle with ratios of 5.5:1. Obviously, there is a tremendous opportunity for improvement through higher compression-ratios which will be permitted by the better fuels without the

necessity for increasing the engine weight per horsepower.

In conclusion, it appears that we cannot predict with any certainty whether the liquid-cooled in-line engine or the air-cooled type will be favored for the larger power-units. From the history of aviation it is safe to say that, whichever type becomes standard in the larger powers, the weight of the powerplant per horsepower will be comparable with that of the air-cooled radial unless a gain in performance is secured because of reduced engine-drag sufficient to compensate for its additional weight. In aviation, we cannot afford to take any chances. Consequently, the standardization of new types is bound to take considerable time both for thorough development and for intensive service testing to prove their reliability.

THE DISCUSSION

R. W. A. BREWER[2]:—The important point to me is in the last paragraph of the paper, wherein Mr. Mead says: "The weight of the powerplant per horsepower will be comparable with that of the air-cooled radial unless a gain in performance is secured because of reduced engine-drag sufficient to compensate for its additional weight."

This point has not yet had full discussion at the meetings of the Society, although it has been discussed at Langley Field for the last two years. As to the different dispositions of engines, our hopes have been rather modified because of the work that has been done on cowling the radial engine and the extensive tests made on the amount of drag of the modified types of cowling. Prior to that, we also had hopes of a different disposition of cylinders in an engine of a type that probably could be installed in the wings so that its parasitic drag would not be an addition to that of the plane as a whole. In view of the Langley Field figures, it seems that there would be a certain amount of additional weight permissible, but that it has not been specified definitely per horsepower. Such figures would be a guide to those who are working on different types of engine. If we could get a permissible pound or fraction of a pound per horsepower as a result of eliminating engine drag, we would like to know approximately what that figure is.

CHAIRMAN G. W. DUNHAM[3]:—If the proper cooling can be developed, perhaps there may be a possibility of a horizontally positioned engine entirely enclosed in the wing.

S. A. MOSS[4]:—Regarding the use of supercharging in decreasing engine weight per horsepower, a Diesel motorship which I visited recently had its power increased about 40 per cent by means of exhaust-gas-driven superchargers. The Diesel-powered ships now in transoceanic service would hardly be possible if it were not for supercharging. At present, the larger radial engines for aircraft increase their power perhaps 20 to 25 per cent by supercharging. Diesel engines for aircraft, such as the Packard, could have their horsepower increased and their weight per horsepower decreased by means of superchargers. I anticipate that progress will be made in decreasing weights per horsepower by a considerable increase in the amount of supercharging.

G. W. DUNHAM

MR. BREWER:—It is scarcely supercharging *per se* which effects the increase in performance of the Diesel engine. It is due to the fact that so much air can be injected into the working cylinder. An inherent property of the Diesel-engine cylinder is that a greater volume of air must be pumped into the cylinder than in the case of the Otto-cycle engine because the former must have an excess of air. In pumping in the ordinary way by the displacer method, one is limited by the displacer rather than by the working cylinder of the engine. The effect of the supercharger is to put a 25 to 30 per cent excess of air into the actual working volume of the cylinder.

CHAIRMAN DUNHAM:—It seems that much can be done with superchargers, especially if the higher-speed engines are considered.

HERBERT CHASE[5]:—Now that we have efficient turboblowers for scavenging and for filling the cylinder, it seems to me that the two-stroke cycle offers possibilities that have not as yet been realized for aircraft engines. There is a great deal of foolish prejudice against two-stroke engines, resulting largely from the failures of this type in early automobiles. The fact is, however, that the faults then encountered have been overcome by engineers who have taken the trouble to find the reason for them. It remains to be seen whether as high a power-weight ratio and as high fuel-economy can be secured with two-stroke engines suitable for aircraft as is obtained from present four-stroke engines; but the simplicity and greater number of power impulses per revolution are factors of importance and should not be forgotten.

A promising step in this direction has been taken in the Junkers Deisel engine which is a two-stroke engine with opposed pistons in the same cylinder. The information I have indicates that this company has built a six-cylinder engine of this type having vertical cylinders in-line. One crankshaft is at the top and the second is at the bottom of the engine, the two being connected by a train of gears, to one of which the propeller is connected. Although involving a rather cumbersome design, this engine is said to develop about 700 hp. and to weigh about 2.8 lb. per hp., the maximum power being developed at 1700 r.p.m.

A. C. HEWITT[6]:—An analysis of the power demands of an airplane would, I believe, show that only the power from 75 per cent upward is used. The only other

[2] M.S.A.E.—Consulting engineer, Aircraft Engine & Accessory Development Corp., Jenkintown, Pa.

[3] M.S.A.E.—Consulting engineer, New York City.

[4] Thomson Research Laboratory, General Electric Co., Lynn, Mass.

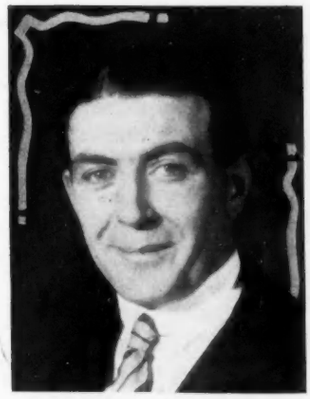
[5] M.S.A.E.—Associate editor, *American Machinist* and *Product Engineering*, New York City.

[6] M.S.A.E.—Engineer, Wright Aeronautical Corp., Paterson, N. J.

requisite of the engine is that it shall idle. The two-stroke engine meets that demand almost perfectly. The limitation lies in the cooling and particularly in the cooling of the piston-head; but, with modern supercharging—by getting some cooling from the supercharged air and more cooling effect from the oil—a considerable range of possibilities is offered and also a chance of building a much smaller engine and of constructing an engine having fewer parts than is now the case. It eliminates one of the difficulties; that is, trouble with the exhaust valves.

MR. BREWER:—Air cooling has shown us without doubt that engines can be run very much hotter than had ever been supposed. It seems to me that liquid cooling, such as the use of a commercial coolant, is only one means of running liquid-cooled engines hotter than they were formerly run; but, we now know, that this can be done.

CHAIRMAN DUNHAM:—Air cooling is a function of the contact between the cooling medium and the part to be cooled. It is a matter of intimate contact. If it is possible to improve that condition by the application of the cooling medium throughout the surface of the part to be cooled, an improvement should be possible in that direction. Liquid cooling is a much more advantageous method from the viewpoint of such intimate contact, as has been demonstrated frequently.



C. H. BIDDLECOMBE

C. H. BIDDLECOMBE[7]:—Reliability is really the most important of all the points mentioned, and the reliability factor is the one factor that seems to me to be at present absent in the liquid-cooled engine. So far as one can determine, the liquid-cooled engine undoubtedly offers certain advantages from the purely theoretical viewpoint. Compared with water cooling, a certain amount of weight is eliminated; but weight is added when a comparison is made with air cooling. Higher efficiency can be obtained inasmuch as the engine can be run hotter, but, as was mentioned, an oil-cooling medium must be used to keep the oil from breaking down under those high temperatures. Due to increased heat, more power is obtained; but, that increased heat again leads to the necessity for using special fuels to prevent detonation and its consequent extensive number of troubles.

On the whole, it seems that the air-cooled engine is really the engine that will endure in developing commercial aviation; because, although the liquid-cooled and the water-cooled engine bears the names "liquid" and "water-cooled," they in reality are only air cooled. The more I see of the way in which liquid-cooled engines are behaving, the more unnecessary they seem in the commercial field, although they seem suitable for racing and perhaps for certain military purposes.

Why should we go back to a principle that has almost been forgotten in commercial aeronautics and carry a quantity of water in the engine to cool that water by air when, by utilizing research and judgment, the engine can be cooled just as readily directly by air without introducing an expensive liquid coolant, without using more expensive fuels such as the anti-detonating fuels, and without introducing radiators which either produce excessive drag or, if they are built into the wing, inherent unreliability? In other words, why return to the subject of introducing a secondary medium between the engine and the air when we have demonstrated very successfully in the last nine years that those mediums are not actually necessary and that the additional power they give is offset by the lack of reliability that is inherent in their use?

MR. HEWITT:—The limitation today lies in the fuels that we have available and it seems to me that we are increasing the compression ratios to secure better fuel-economy faster than the oil companies can supply us with better fuels, and that this is true also of the liquid-cooled engine. Of course, there is a slightly better margin because we are eliminating some of the hot-spots, but when the hot-spots are eliminated with liquid cooling the tendency will be to raise the compression ratio higher, either by changing the compression ratio of the piston or using a higher supercharging-ratio. While liquid cooling offers this chance of decreasing the number of hot-spots, we must guard against going ahead of the limitation in present-day commercially available fuels.

T. E. TILLINGHAST[8]:—There is much truth in what Mr. Biddlecombe says to the effect that possibly, in considering high-temperature cooling, we are looking toward some type of engine which may have less reliability. However, other things being equal, one factor that is favorable toward liquid cooling is that we can assume that a liquid-cooled engine will withstand more abuse from bad fuels than will an air-cooled engine. With a liquid surrounding the cylinder-wall, this provides a definite limit to the temperature. Under some conditions of bad detonation, an air-cooled engine might give trouble; but a liquid-cooled type of engine might continue to operate successfully under those conditions.

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[7] Faculty lecturer on air transport at New York University, New York City.

[8] M.S.A.E.—Executive engineer, The Pratt & Whitney Aircraft Co., Hartford, Conn.

Geared Centrifugal Superchargers for Airplane Engines

By SANFORD A. MOSS[1]

DETROIT AERONAUTIC MEETING PAPER — *Illustrated with* PHOTOGRAPHS

FOUR independent purposes for the geared centrifugal supercharger, now in extensive use on commercial airplane engines, are listed by the author, who states that these are actually centrifugal compressors geared from the engine crankshaft and having such rotative speed and design detail as to produce an appreciable pressure-rise. A system of cooperation between the engine and the supercharger builders has been devised so that the supercharger is an integral part of the engine. The high-speed impeller is the only part that the supercharger manufacturer supplies. The system enables the engine builder to have the benefit of a supercharger carefully designed and superintended by specialists in the art.

After describing the centrifugal compressor, which builds up at the diffuser exit a pressure appreciably greater than that at the impeller exit because of the conversion of velocity into pressure, the author mentions some details of the history of supercharger development and comments upon the design of the built-in supercharger.

Stating that the carbureter arrangement in supercharged engines is simple, there being but a single path from the atmosphere to the inlet of the supercharger impeller, the author mentions the benefit obtained by using rotative rather than reciprocating devices as regards a tendency toward compact design, as well as the higher speeds attainable by moving parts that are perfectly balanced rotating wheels, and remarks also upon the importance of selecting high-quality material and maintaining adequate inspection.

As used in airplane engines, one of the most important advantages of the supercharger is the improvement in distribution, according to the author, who enlarges upon this subject and discusses also the theory of supercharging an engine at sea-level, inclusive of the power limit for supercharging. Other subjects treated include supercharging at altitude, the maintenance of carbureter efficiency, and speed variation with a supercharged engine. In conclusion, the author remarks that any good engine not equipped with a supercharger will be a better engine if a supercharger is installed.

THE GEARED centrifugal-supercharger, now in extensive use on commercial airplane-engines, has never been adequately presented in print. Such equipment is used for some or all of four independent purposes:

(1) For improvement in distribution of the engine charge, with carbureter and inlet-pipe simplification
(2) For maintenance of full power at moderate altitudes
(3) For increase of power at sea-level for short periods and emergencies
(4) For increase of normal full-load power at sea-level.

The superchargers in commercial use are centrifugal compressors geared from the engine crankshaft and having such rotative speed and design detail as to produce an appreciable pressure-rise. A system of cooperation between the engine builder and the supercharger manufacturer has been devised so that the supercharger is not an external accessory but is an integral part of the engine.

As shown in Fig. 1, the air passages, impeller casings and diffuser are built as integral parts of the engine crankcase according to designs furnished by the supercharger manufacturer; hence the high-speed impeller itself becomes the only part which the supercharger manufacturer supplies. This system enables the engine builder to have, with a trivial addition to the weight, cost and size of the engine, the benefit of a supercharger carefully designed and superintended by specialists in the art.

The Centrifugal Compressor Described

The centrifugal compressor is an apparatus for compressing air to appreciable pressures in a similar way to that of the fan blower and the centrifugal pump; but the various air and impeller speeds and the pressure rise produced in proportion to the density are very much greater than in such blowers and pumps. Many theoretical and experimental data have been accumulated in the United States since 1904. This has made possible the construction of a particular type of centrifugal compressor having very high efficiency.

In the centrifugal compressor—and supercharger—the air enters the high-speed impeller near the center, through an eye, or annulus, proportioned to give a suitable compromise between too high a velocity of the fluid as it enters the impeller and too high a velocity of the blade tips which receive the fluid. The inlet tips of the impeller blades are bent forward so as to receive the fluid without shock. This is because the fluid at the inlet edges of the blades is proceeding at a certain angle *relative to the blades*. This relative direction is, of course, different from the absolute direction of the inlet fluid in space. The inlet edges of the blades must be exactly faced in the relative direction of the fluid. The impeller passage then curves so as to bend the

[1] Thomson Research Laboratory, General Electric Co., West Lynn, Mass.

stream of the fluid gently and pass it to the exit edges of the impeller blades, which are always in a radial direction.

During the flow along the impeller, from the impeller entrance to its exit, the fluid has a certain relative velocity with respect to the impeller. The passage areas are maintained so as to keep this relative velocity comparatively small to minimize friction loss. At the impeller exit, an appreciable pressure has been produced as a result of the so-called "centrifugal effect." Further, the fluid has a high absolute velocity which is practically the velocity of the tips of the impeller blades, a velocity representing a certain amount of kinetic energy which has been imparted to the fluid but is not in useful form. This high-velocity fluid is discharged into a carefully designed passage beyond the impeller, which is called a "diffuser" and is shaped so as to slow down gradually the velocity of the fluid with the minimum of friction and eddy loss. Such an efficient decrease of velocity results in an increase of pressure; hence, at the diffuser exit, a pressure appreciably greater than that at the impeller exit exists because of the *conversion of velocity into pressure.*

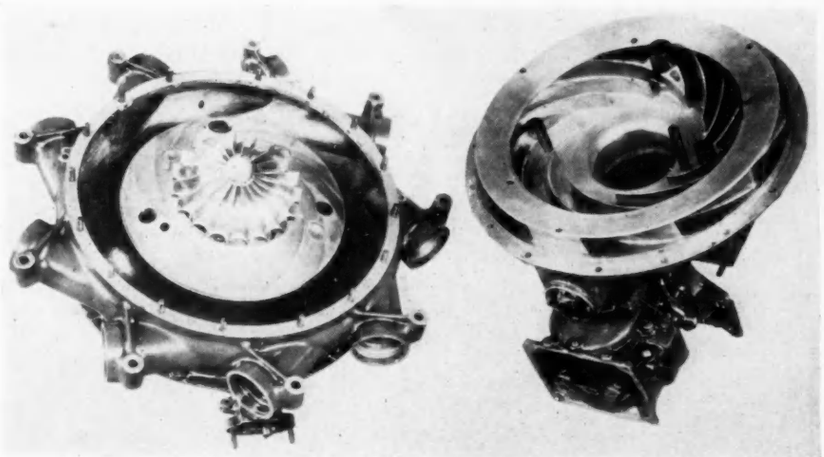

FIG. 1—BUILT-IN GEARED CENTRIFUGAL SUPERCHARGER USED ON THE PRATT & WHITNEY WASP ENGINE

Centrifugal compressors of this type are used for many commercial purposes which require air at pressures of from 1 to 30 lb. per sq. in., as in foundry cupolas, blast furnaces for pig iron, pneumatic-tube and conveyor systems, air supply for furnaces, exhaustion and compression of gas in illuminating-gas plants, and the like. Fig. 2 shows commercial machines of various types and gives an idea of the state of the art of which supercharger design is a part.

Centrifugal compressors operate at peripheral speeds ranging from 500 to 1500 ft. per sec. at the tip of the impeller. In present airplane-engine superchargers, the diameters are such as to require rotative speeds of 15,000 to 30,000 r.p.m. The air and the impeller speed are such as to make it imperative that the passages, blade angles and other items of mathematical design be attended to so that the high-speed flow is handled without shock or loss. When this is done the centrifugal compressor furnishes a means not equalled by any other device for compressing air over a wide range of conditions within the pressure range considered. It is easy to make an improper design, which may have any low efficiency whatever.

Development of Airplane Supercharging

Many proposals for the supercharging of airplane engines were made during the war and it was natural that both the General Electric Co. and the affiliated British Thomson-Houston Co. should be called upon for development work. This was greatly facilitated in the United States by the cooperation of the engineers of the Army Air Corps, the Navy Bureau of Aeronautics, the National Advisory Committee for Aeronautics and the engineers of the Curtiss, Wright and Pratt & Whitney companies. The development of the exhaust-driven supercharger proceeded most rapidly. It comprises a centrifugal compressor having a general design identical with that of those described in this paper, but is driven by turbine wheels operated from the engine exhaust-gases and is primarily suited for flights at altitudes of 20,000 ft. and upward.

Fig. 3 shows one of the first geared superchargers; it was built for the Curtiss company in 1917. The development work was continued with similar designs on various other engines, and many test flights were made. During all of the early work, water-cooled in-line engines were used and the supercharger was attached by gearing to the crankshaft in various ways. The difficulties encountered with some of the early gear-layouts have been eliminated, and a number of in-line engines equipped with geared centrifugal superchargers, such as the Wright inverted V-type shown in Fig 4 (*c*), are now operating successfully. All of the original geared superchargers were separate units bolted onto the engine and required a separate gear-train and an external gear-casing, a supercharger casing, and a set of air conduits.

Design of the Built-In Supercharger

In the development of the radial air-cooled engine, it was proposed in both England and the United States to use the centrifugal supercharger directly built into the engine without external casings. This system is now in extensive use. It requires that the engine be specially designed so that the supercharger fits properly. Some of the engines illustrated herewith were expressly designed from the beginning to secure the benefits of supercharging. In other words, engines initially operated without superchargers have been redesigned for the addition of a supercharger, which has proved to be a comparatively easy matter.

In these supercharged engines, a group of gears serves for the magnetos, starter, oil-pump and generator, and an additional gear or two for the supercharger is connected with this train, the impeller and gear train in a typical case being shown in Fig. 5. In most cases, two pairs of gears are used to get the complete supercharger gear-ratio, which varies between 7 to 1 and 14 to 1.

The supercharger impeller is designed to have ex-

ceedingly low inertia; but the acceleration and deceleration of the impellers, as the engine is started and stopped, involves some gear stresses. The gears easily withstand this stress in most cases, but, in some cases, for the higher gear-ratios, it is thought desirable to install a few discs and a spring which constitute an automatic slipping clutch which is set to operate without slipping during full-speed service but to slip somewhat during starting and stopping.

The casing in which the impeller rotates and the diffuser beyond the impeller are formed by the engine castings. This has involved no difficulty in the engine design and comparatively little increase in weight. The diffuser leads into an annular passage from which are led pipes to the inlet valves of the various cylinders. Fig. 1 shows an exploded and typical view of the supercharger arrangement originally used in the Pratt & Whitney Wasp and Hornet engines. Diffusers are made with and without vanes.

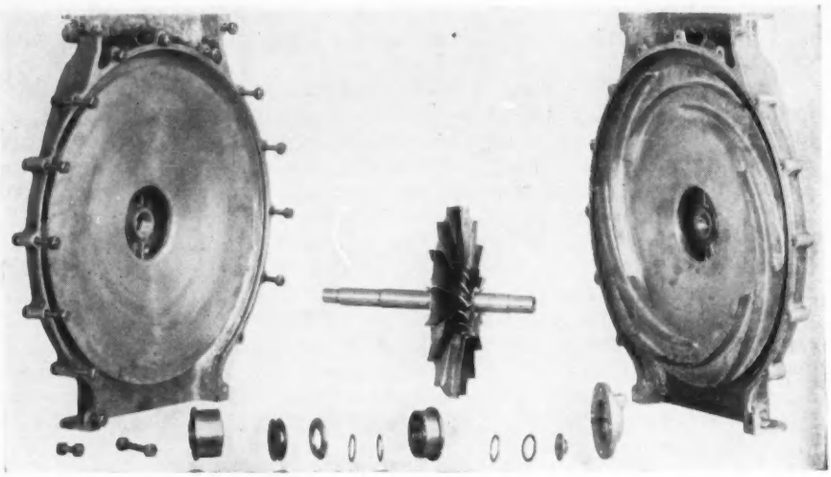

FIG. 3—ONE OF THE FIRST GEARED SUPERCHARGERS DESIGNED AND BUILT FOR A CURTISS ENGINE BY THE GENERAL ELECTRIC CO.

Carbureter and Other Considerations

The carbureter arrangement in supercharged engines is very simple, there being but a single path from the atmosphere to the inlet of the supercharger impeller. A single-barrel carbureter is used in most cases; but when the diameter of a single barrel to handle the air volume would be unduly great, a double-barrel carbureter is used and the two barrels are directly connected by a common inlet and discharge, thereby eliminating the complication of separate paths through independent carbureter barrels to separate banks of engine cylinders, as often occurs with unsupercharged engines. It usually is desirable to have a hot-spot or heater connected with the exhaust, between the carbureter and the supercharger inlet, to avoid difficulty with ice formed in cold weather from the gasoline vaporization. The ordinary type of carbureter throttle is used in the passage between the carbureter and the supercharger inlet, and this furnishes the entire supercharger regulating arrangement.

The engineering divisions of the engine manufacturer and the supercharger manufacturer cooperate in making the layout of a supercharger that is to be built into an engine. Successive changes in the design are made until one is finally achieved that fits into the engine easily and gives minimum expense in manufacture, but which operates with adequate efficiency from the viewpoint of the supercharger specialists, who take the entire responsibility for the air passages from carbureter inlet to inlet-valve housing.

A characteristic which has enabled turbine machinery of all kinds to displace reciprocating machinery is the comparative size for a given duty. Steam turbines have eliminated reciprocating steam-engines as prime movers for powerplants, because their weight and space requirements are less. The centrifugal compressor shares this characteristic of turbine apparatus and provides a less bulky unit with which to compress a given amount of air, particularly when the centrifugal compressor is built into the engine.

One of the essential characteristics of turbine machinery, including the centrifugal compressor, which tends toward compact design is the fact that the only moving parts are perfectly balanced rotating wheels and much higher speeds are possible than with reciprocating machinery. It is imperative that exact knowledge be available for computation of the stresses set up by the speeds used and that the rotating parts be designed to withstand the stresses with entire safety.

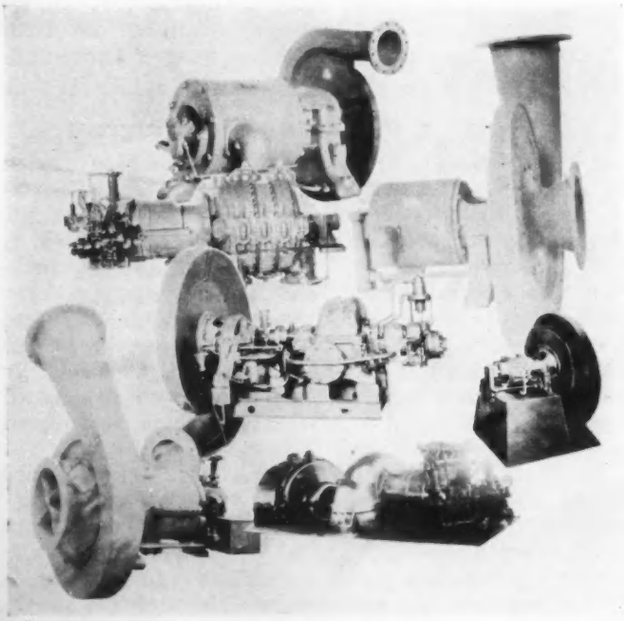

FIG. 2—TYPES OF CENTRIFUGAL COMPRESSORS IN COMMERCIAL USE

Superior Material and Inspection Required

The selection of material, its heat-treatment and inspection must be attended to carefully, so that the high rotative speed of the supercharger can give an important advantage. Airplane supercharger impellers con-

structed under the supervision of a capable staff of mathematicians, metallurgists and engineers have never failed in service.

The material used for the impellers illustrated herewith is a heat-treated forging of aluminum alloy. The impellers are forged as nearly as possible into final shape and milled out of this solid forging by special machinery so as to give proper thickness of blade. The heat-treating and aging processes are interspersed with the manufacturing processes in the most convenient way.

Supercharging Improves Distribution

One of the most important advantages of the supercharger as used in airplane engines is the improvement in mixture distribution. Various devices have been used to secure good distribution and vaporization, but none have had the success of the supercharger illustrated in Fig. 1. Some of the other devices have been to use various forms of tortuous passages that were supposed to provide a satisfactory mixture, to use a small carbureter-diameter that gave a high velocity but entailed loss of power due to choke, to use a large impeller at a very slow speed obtained by installing it directly on the crankshaft, and to use several sets of intake passages through independent carbureter barrels. But none has given the smoothness of operation and uniformity of power in all clylinders that is secured by the centrifugal supercharger.

In some early cases a centrifugal supercharger was used primarily for its effect in improvement of distribution, and this was called "rotary induction." But an efficient supercharger operating at a speed high enough to give the proper improvement in distribution

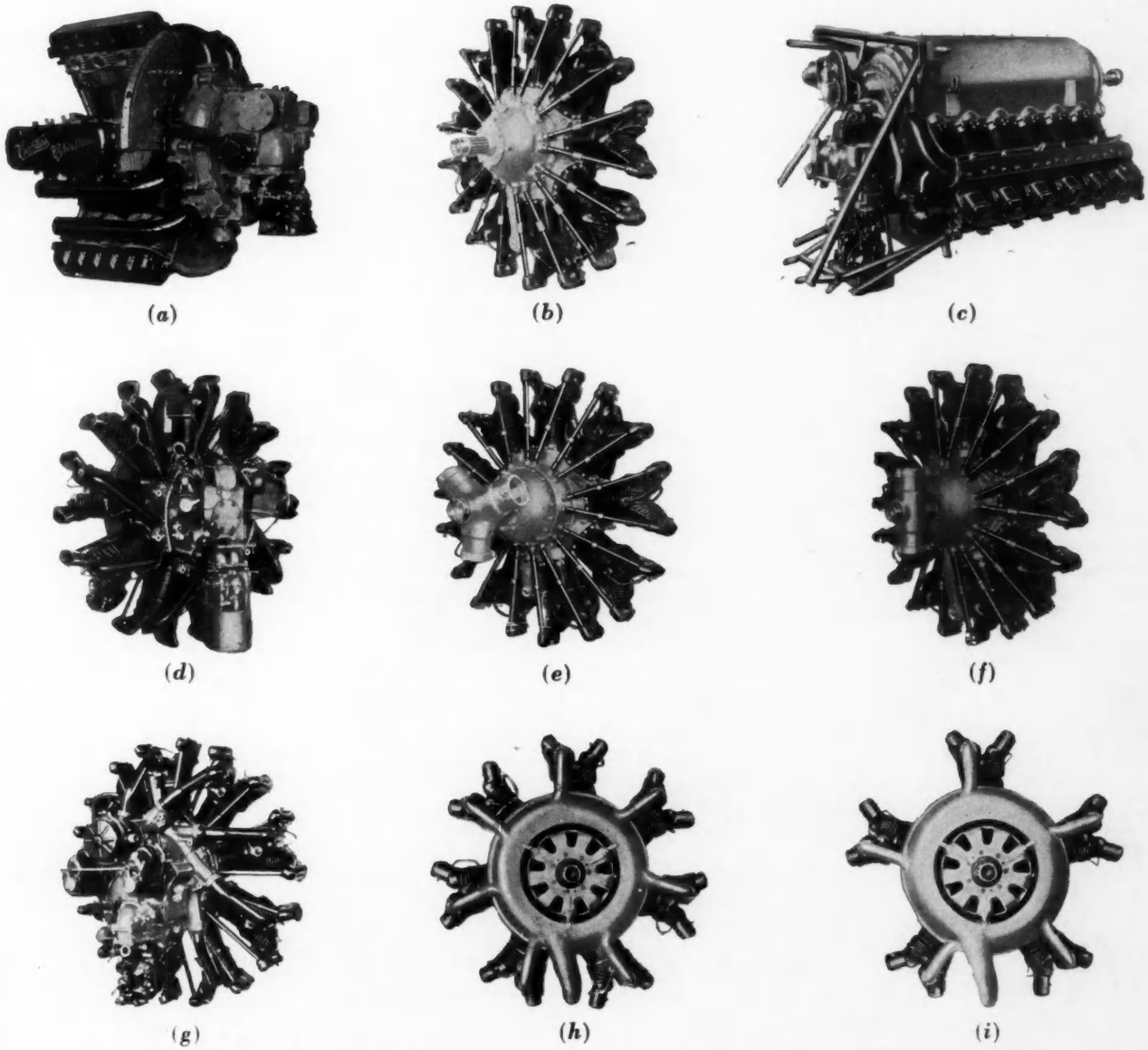

FIG. 4—SOME OF THE COMMERCIAL AIRPLANE-ENGINES REGULARLY EQUIPPED WITH A GEARED SUPERCHARGER

(*a*) Curtiss Chieftain, 600 Hp.; (*b*) Pratt & Whitney Hornet, 575 Hp.; (*c*) Wright Inverted-V, 550 Hp.; (*d*) Wright Cyclone, 525 Hp.; (*e*) Pratt & Whitney Hornet, 500 Hp., and (*f*) Wasp, 420 Hp.; (*g*) Wright Whirlwind, 300 Hp., (*h*) 225 Hp., and (*i*) 165 Hp.

also gives a positive pressure in the intake manifold of about 3½ in. of mercury, or an absolute pressure of about 33 in. of mercury. If no supercharger were used, a negative pressure of about 1½ in. of mercury, or an absolute pressure of 28 in. of mercury, would exist in the manifold. This represents a percentage increase in charge of 5/28, or about 18 per cent, corresponding to a power increase at full throttle of nearly the same amount. This is an appreciable amount of supercharging; hence, there is no reason for calling the unit a rotary inductor rather than a supercharger.

Some attempts have been made to use an impeller of low efficiency without a suitable diffuser and with such design as would result in eddy and other losses. The idea was that the temperature rise given by such design would improve the vaporization, but any temperature rise increases the volume of the charge and decreases the weight of the charge filling a given cylinder. Further, completely satisfactory vaporization and distribution are attained by use of a supercharger of maximum efficiency and minimum temperature-rise. There is, therefore, no meaning to "rotary induction" in itself, and the thing accomplished is the improvement in distribution which inevitably accompanies the use of an efficient centrifugal supercharger, aside from the direct effect of supercharging.

It is believed that the diffuser exercises an important function in the vaporization so successfully accomplished by an efficient supercharger. In order that a diffuser may exercise the function for which it is primarily provided, the conversion of velocity into pressure, the air is necessarily guided through it in curved paths with gradual decrease in velocity. The drops of gasoline that leave the forward edge of each of the impeller blades are discharged with constant velocity and in straight lines into the air passing along in the diffuser. The result is that each gasoline drop continually meets new air, so that complete vaporization is secured by the time the end of the diffuser is reached. The compression of air by a centrifugal supercharger also creates a slight temperature rise, and this gives a slight increase in the vaporization. But with ordinary amounts of supercharging, the temperature rise is not sufficient to give appreciable decrease in the weight of cylinder charge.

The vaporization system mentioned is sufficient without the addition of hot-spots or heat of any kind for ordinary weather, but some sort of passage heated by the exhaust must be added to avoid clogging of the carbureter by ice formed by the vaporization of gasoline in cold weather.

Theory of Supercharging at Sea-Level

In general, the power of an engine having a given displacement varies directly with the absolute pressure in the inlet pipe, but if the supercharger pressure or efficiency is such as to give appreciable temperature-rise, there is decrease of charge due to this effect and then the power varies approximately as the density of the entering charge.

It is assumed that, as the amount of supercharging is increased, the compression pressure is maintained. This requires an increase of the clearance volume. In some cases, by using specially prepared fuel, supercharger pressures have been increased without changing the clearance volume, which gives increased compression-pressure and results in an additional increase in power over that caused by the direct increase of charge on account of supercharging. With suitable valve timing, a supercharged engine may have the clearance space scavenged and filled with fresh charge. In this event, the power increase is greater than that corresponding merely to filling the displacement with increased charge.

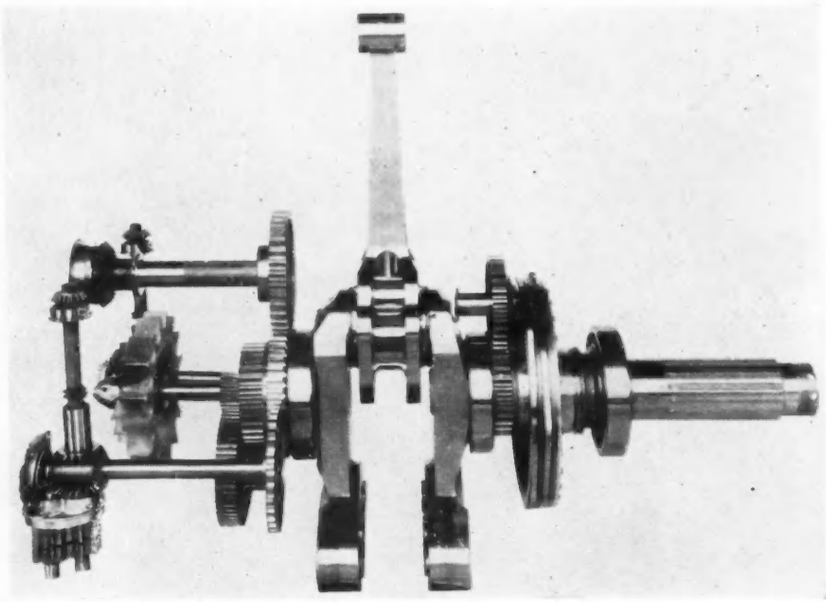

FIG. 5—PRATT & WHITNEY WASP-ENGINE IMPELLER AND PART OF ITS GEAR TRAIN

A supercharged engine with a given compression-pressure does not finally expand the charge to as low an absolute pressure as an unsupercharged engine with the same compression pressure. There is therefore a theoretical loss of a small portion of the toe of the indicator card which tends to increase the fuel consumption. But other effects equal or counterbalance this, such as the decreased percentage of friction loss on account of the increase of power from a given cylinder and, for the same reason, the decreased percentage of jacket water or air-cooling loss. Many test data indicate as a net result that the fuel consumption with a supercharged engine is equal to, or in some cases lightly less than, that without supercharging, and in no case has a supercharger been found to increase the fuel consumption.

Consideration of the exact circumstances at the toe of the indicator card will show why there is no actual loss of efficiency. There is an appreciable amount of engine friction which acts as "friction pressure," and this is subtractive from the indicator-card pressure. The pressure at the toe of the card when the final ex-

pansion is taking place is small, being but little greater than this friction pressure. The loss caused by lack of expansion represents the small difference. The portion of the compression which occurs in the supercharger replaces compression in the cylinder with equal efficiency. The maximum temperature with supercharging is the same as without, with the same compression pressure; hence the power increase with a given cylinder results in a decreased proportionate loss on account of cylinder cooling. The net result of all of these effects can hardly be computed theoretically but ample test evidence shows that there is no net loss in specific fuel-consumption.

Power Limit for Supercharging at Sea-Level

In all cases it has been easy to supply a supercharger to increase the power of an engine at sea level to the maximum amount advisable, from the viewpoint of the engine. Sometimes the limit is detonation, and this limit can be increased by the use of antiknock or by increase of clearance volume. At other times the limit is the maximum power which the engine will deliver with safety, so far as durability is concerned. Engine builders are making gains in this connection and the safe values of engine power are rising. However, some engines will operate for long periods without overhauling if the maximum power is limited and will require more frequent overhauling if this limit is exceeded. Such engines may use superchargers with capacity for emergency or altitude operation and with limitation of the amount of supercharging for normal operation. At times the limit of supercharging is governed by temperature rise of the supercharged air on account of compression to high supercharger-pressures. In this case the limit can be increased by use of an air-cooler. The limit of supercharging at sea level is sometimes determined by the temperature attained by the cylinder with the increased power and the available amount of cooling.

In the case of radial air-cooled engines of the smaller sizes, the supercharging limit has never been reached. As the size of the cylinder decreases, the ratio of surface to volume increases; therefore the cylinder can radiate a greater amount of heat in proportion. This means that the mean effective pressures of the smaller engines can be increased by using large amounts of supercharging and still maintain safe temperatures. A small radial air-cooled engine with a supercharger thus added to give the maximum desirable power and still maintain safe temperatures would have a much less cost and weight per horsepower than an unsupercharged engine. Therefore a great possibility of progress exists in the supercharging of small engines.

Supercharging at Altitude

It is comparatively easy to supercharge an engine so that it will maintain full sea-level power at altitudes up to about 12,000 ft. Supercharging for greater altitudes usually requires the use of an air-cooler. If a supercharger is added to an engine primarily for the purpose of increasing power at altitude, the amount of power developed at sea-level is an independent consideration. Very often, owing to some one of the foregoing limitations, the engine will not safely deliver the amount of power corresponding to full supercharging at sea level. In such case, the carbureter throttle-valve must be adequately regulated and must not be opened wide until considerable altitude is attained. Sometimes a throttle stop is provided which limits the opening when near sea-level; however, by making a certain extra motion, the throttle valve can be moved past the stop, and this is permissible after some altitude is reached.

Maintenance of Carbureter Efficiency

Without a supercharger, the power of an airplane engine falls off rapidly even at moderate altitudes. The restoration of sea-level power up to altitudes of about 12,000 ft., which is easily possible with a geared centrifugal supercharger, gives a remarkable improvement in the performance of the airplane, often increasing the speed 50 per cent at altitude, compared with the speed without a supercharger.

In a number of cases a given supercharged engine has inadvertently been produced with a restricted inlet or too small a carbureter, so that its full power was not delivered. It has been comparatively easy to change the gear ratio of the supercharger so as to speed it up somewhat and thus create an additional pressure-rise to overcome any inadvertent friction and obtain full power. This practice is not to be commended and is to be used only as a makeshift. Every effort should be made to have all passages so large that the friction drop will be negligible when a supercharger is used.

It is possible to go much farther in this connection with a supercharger than without, because, when no supercharger is used, certain velocities must be maintained in the inlet passages and through the carbureter choke to give satisfactory vaporization and avoid having gasoline drop out of the mixture. With a supercharger, no attention need be paid to such matters; the velocities may be comparatively low and the choke comparatively large, regardless of gasoline vaporization, because the vaporization secured by the supercharger alone is entirely adequate under these circumstances. For this reason an engine with a supercharger can have a larger choke and larger inlet passages than otherwise, so that the total pressure-increase at the inlet valves is greater than the pressure rise of the supercharger by the decrease in inlet friction.

Speed Variation with a Supercharged Engine

Normally, an airplane engine is not operated through a very wide range of speeds, but it is frequently desirable to operate for a period at speeds above normal. Any unsupercharged aviation engine usually delivers its maximum power at a given speed, with less power before and after. The reason is that, as the engine speed increases, the drop through the carbureter, inlet pipes, and inlet valves due to friction increases so that there is a successively decreasing amount of charge. This decreases the torque and the power. If it were not for this additional loss caused by friction at increased speed, the torque would be constant and the power would increase directly with the speed. Actually, the torque successively decreases; so the power at first increases at a slower rate than the speed increases, presently reaches the maximum, and thereafter decreases instead of increases on account of speed. This possibility is eliminated with a geared supercharger.

Such a supercharger might be designed to exactly overcome the drop due to friction in the carbureter, inlet passages and inlet valves, so as to give exactly atmospheric pressure in the cylinders at the beginning

(*Concluded on p.* 160)

Reasons for the 16-Cylinder V-Type Cadillac Engine

Detroit Section Paper

By W. R. STRICKLAND[1]

Illustrated with Charts, Photographs and Drawings

HEREIN the author reviews the engineering considerations that led to the adoption of the 16-cylinder V engine as the next step after the 8-cylinder V-type. He lists the demands for increased power and states that, to meet the conditions adequately, a 40-per cent increase was required. Objections to increasing the diameter of the cylinder bore are cited; higher mean effective pressure cannot be used unless it does not interfere with smooth operation, especially at low speeds; changes in design could not be counted on to improve thermal efficiency; supercharging was not believed to be practical; and the four-speed transmission seemed to offer no solution.

By the process of elimination, an increase in the number of cylinders was the solution arrived at, and charts are presented to show the greater smoothness of operation of the V-16 compared with the 6 and 8-cylinder engines. Advantages of using a smaller bore and piston displacement per cylinder are pointed out; the reason for disposing the two banks of 8 cylinders at a 45-deg. angle is given, and the general layout and some of the details of design are described.

In conclusion, the author lists, as the results obtained, "silkiness" of operation, with smooth and quiet pick-up, running on the road and over-running; satisfactory operation with a 5.5:1 compression-ratio; rapid acceleration and a top speed of more than 80 m.p.h. with a seven-passenger sedan; and fuel economy at medium car-speeds that equals that of the V-8 engine.

WITH the continual development in smoothness and power of the Cadillac 8-cylinder V-type engine went the research work of the engineer into greater and finer attainments for the future. The advantages of the 8-cylinder engine had been exemplified first in the V-type and then, by others, in the single block. Great advance was being made in details, but, further than that, improvement called for a survey of the possibilities of advancement in engineering design. The survey of possibilities had to be preceded or paralleled by a survey of the demands of the more enlightened engineers and car owners. The path of advancement that was considered then was along the following lines of improvement:

Rapid acceleration
High speed
Smooth operation
Quiet operation
Ease of operation
Ease of quick deceleration
Character and beauty of appearance
Class and finish of appointments
Riding comfort.

Nearly all of these would influence the engine design, but other problems to be met, which were of even greater demand, were weight increase and wind resistance. Cars in this class have increased in weight to well over 5000 lb., and, with custom bodies, wire wheels, fender wells, trunk racks and other special equipment, 6000 lb. is exceeded. The wider three-passenger seats increase the power rquirements at high speed.

A car of these superior qualities, with the attendant weight, which is an assurance of sturdiness, long life, comfort and safety, must accelerate in traffic with or ahead of the jam, must take all main roads over hills or mountains with ease and at highest permissible speeds, and attain and maintain high speed on those stretches of the new super-highways which are multiplying rapidly all over the Country.

The demands enumerated called for another increase in power. To meet the conditions adequately would require an additional 40 per cent. How was this to be obtained? Could the piston displacement be increased?

Fifteen years ago the Cadillac set the pace and showed the limitations of the size of cylinders, pistons and other engine parts. At the time of the introduction of the V-type eight the fine cars of America were called 38's, 48's and some 68's, meaning N.A.C.C. rating. The cylinder bores were 4, 4½ and 5 in. The brake horsepower was 70, 85 and 120. What happened?

The many advantages of the 8 and 12-cylinder engines led to complete change of thought. Eights and twelves began to multiply and to displace the old large-bore engines. At present, increasing the displacement of the eight meant going back to larger bores, with the mechanical and dynamical difficulties that were present in the old 48's and which are inherent with large bores, as will be detailed later.

Increase of power by higher mean effective pressure can be of value only if smoothness, especially at low speeds, is not interfered with. Non-detonating fuels or ethyl gasoline are helpful, and higher compressions are the rule today as a result, but thermal improvements from design could not be counted on to give the desired results. Supercharging, except for racing purposes, was not believed practical for the same reasons.

[1] M.S.A.E.—Assistant Chief Engineer, Cadillac Motor Car Co., Detroit.

The so-called four-speed transmission used by some to obtain the effect of increased power can be made so as to improve gearshifting in the ordinary transmission by reducing the steps from low to high gear, but its adherents generally prefer an overdrive with the same steps in the gearshift. The consequence is that it is still a three-speed transmission, as no one uses the low speed, and the car is underpowered.

Solution Afforded by More Cylinders

A solution for the problem of increased power is an increased number of cylinders, 16 for the model under discussion. The reason that 16 cylinders provide a satisfactory solution will be explained.

For many years of Cadillac production the chassis parts, driving train, axles and other components have been attuned to the demand of service upkeep and long life; stresses in the various parts, lubrication and maintenance have been worked down to a satisfactory basis.

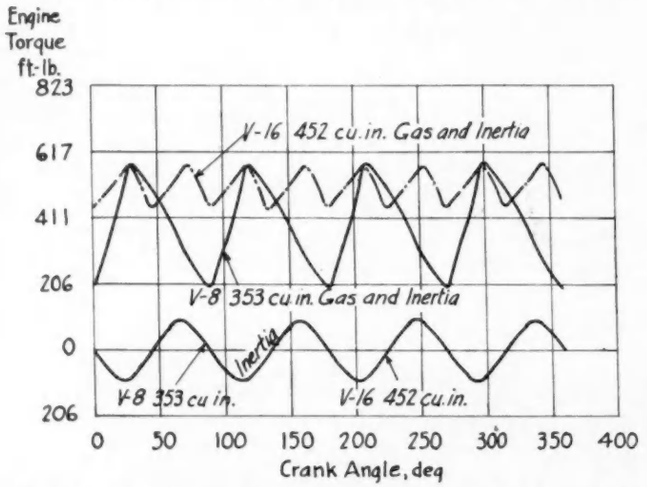

FIG. 1—COMPARISON OF CADILLAC V-8 AND CADILLAC V-16 TORQUE CHARACTERISTICS AT 4200 R.P.M.

The Curve at the Zero Line Shows that the Inertia alone of the Reciprocating and Rotating Parts of the Two Engines Is the Same. The Two Upper Curves, of Combined Gas Torque and Inertia, Show that the Maximum Peak Torque Is no Greater in the Larger 16-Cylinder Engine than in the 8-Cylinder Engine and that the Variations of the Torque Curves Is Very Much Less in the Former

An increase in power, therefore, without raising the stresses or necessitating the designing and introduction of new parts, would be highly desirable.

Fig. 1 is a plot of the combined inertia and gas torque of the 8 and 16-cylinder engines of 353 and 452-cu. in. displacement respectively. It shows two very important things; first, that the maximum peak torque is no greater in the larger 16-cylinder engine than in the 8-cylinder, and, second, that the variation of the torque curves is very much less in the former. This means that, neglecting the flywheel effect, the working parts would have not only less stress but longer life, as the fatigue stresses are less damaging when the variation or stress range is less. Previous experience with 12-cylinder engines provided the facts to validate this theory and to also point the way to smooth operation.

To review the question of smoothness, Fig. 2 shows the inertia diagrams of 6, 8 and 16-cylinder engines, all of the same total piston displacement and detail characteristics, though with the various V angles used or

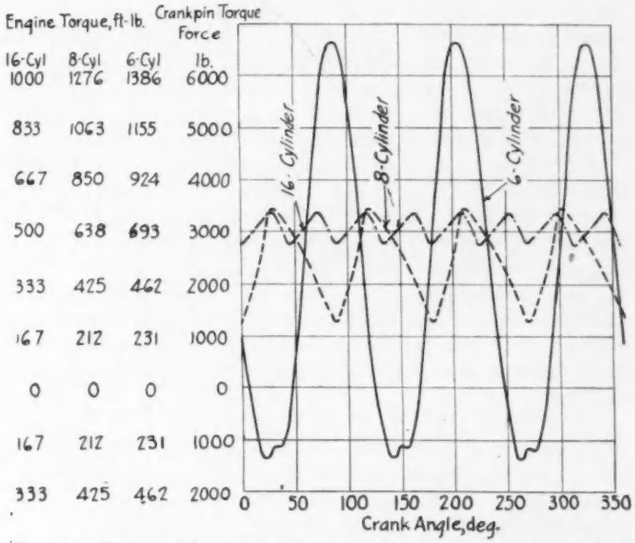

FIG. 2—TORQUE-CHARACTERISTICS, INERTIA ONLY, OF 6, 8 AND 16-CYLINDER V-ENGINES, EACH OF 452-CU. IN. PISTON DISPLACEMENT, AT 4200 R.P.M.

These Inertia Diagrams Show that the V-16 Engine with a 45-Deg. V Gives No Variation in Torque and the 8-Cylinder Engine Gives Very Little Inertia Torque Variation

contemplated. Note that certain V-type engines have the correct angle between the cylinders to give virtually

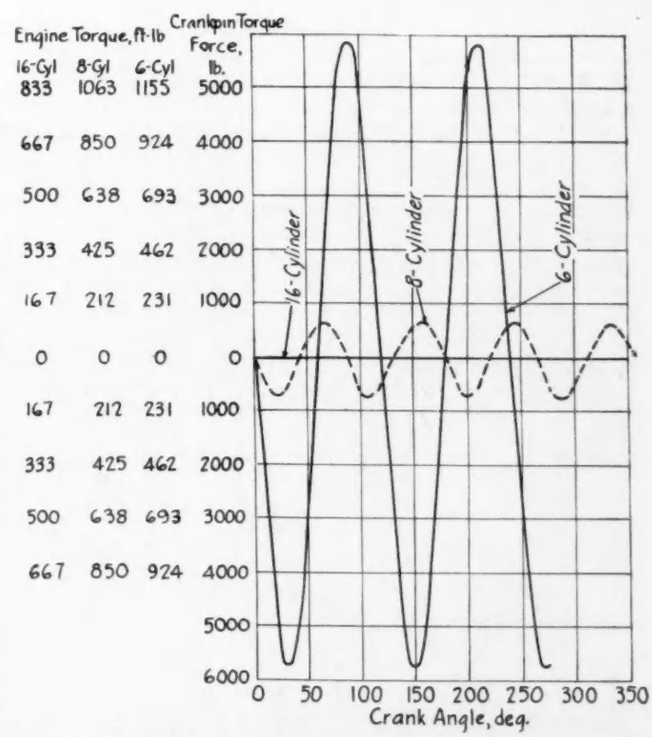

FIG. 3—COMBINED INERTIA AND GAS TORQUE OF SAME THREE ENGINES IN FIG. 2, AT 4200 R.P.M.

This Is the Real Basis of Comparison While Driving a Car, Whereas Fig. 2 Is More Representative of the Performance When Over-Running the Engine, as in Coasting with the Engine Connected through the Clutch. Compare These Curves with Those in Fig. 1, Which Give the Actual Comparison of Combined Gas and Inertia Torque Obtained with the V-16 Engine of 452-Cu. In. Displacement and the V-8 Engine of 353-Cu. In. Displacement

no variation in torque, and that the 16-cylinder V-type 45-deg.-angle engine is ideal. This results in smooth external operation, and, as the pistons, connecting-rods and related parts are smaller in the V-16 engine, the internal stresses are lower and effects which would result from internal stresses are also smoothed out.

Fig. 3 shows the same inertia torque combined with gas torque. This is the real basis of comparison while driving the car, whereas Fig. 2 is more a picture of the performance when over-running the engine as in coasting with the engine connected through the clutch. Fig. 1 gives the actual comparison of combined torque obtained with the designs and sizes of engines under discussion, that is, the Cadillac V-8 and V-16, plotted on an engine-torque basis the peaks of which are the same.

Details Favoring Small-Cylinder Units

Quietness of operation of the present V-8 was regarded as satisfactory, and the same satisfaction was possible even with the increased number of parts in the V-16, especially of valves. But with the decision to use overhead valves, the automatic hydraulic clearance-adjuster was adopted to assure quietness.

Mention was previously made herein of the improbability of securing the desired increase of power by raising the compression. I must add that we did not believe it possible to obtain the increase by any known design of combustion-chamber, either L-head or overhead-valve type, although we have followed with research work all the celebrated suggestions.

Temperature limitations on large-bore engines call a halt in that direction also. Already the various elements in the cylinder are heated to such a temperature that limits are placed on higher compressions and volumetric capacity. Reducing the bore and piston displacement per cylinder by using more cylinders, with the resulting improved cooling conditions, allows a higher compression to be carried on top of a higher volumetric capacity.

Unless the heat absorbed by the piston heads is rapidly conducted to the cylinder-walls, the build-up will cause a hot-spot and loss of power. The amount of heat absorbed varies as the surface or the square of the bore; also, the greater the bore is, the slower the heat travels from the center of the piston head to the piston-rings. Taking the heat absorption of the V-8 as 100 per cent, the 3-in. piston head of the V-16 will absorb only 70 per cent as much heat per piston; therefore, with equal ring-cooling efficiency, the temperature of the center of the piston will be 30 per cent lower.

The amount of heat to be dissipated to the piston, valves, cylinder-walls and so on goes down as the cube of the bore, where comparative designs are used. The area of ring surface to conduct the heat from the piston to the cylinder-wall is relatively greater in the smaller bore, as the circumferential surface drops only directly as the bore. Small valves also are cooler, as the seating or cooling surface is relatively greater for the amount of heat absorbed.

The effect of these four heat-factors: (*a*) less heat per bore unit, (*b*) less absorption of heat by the piston-

FIG. 4—CRANKSHAFT OF THE 16-CYLINDER 45-DEG. V-ENGINE

Compactness of This Engine Gives the Minimum of Crankshaft Length. Main Bearings Are 2⅝ In. in Diameter and the Throw Bearings Are 2½ In. in Diameter, Giving an Overlap of ⅛ In. The Counterweights Are Formed Integrally with the Crankshaft Cheeks. The Torsional Vibration Is Very Small and Requires only a Small Harmonic Balancer in Front, Combined with the Fan Pulley, To Eliminate It

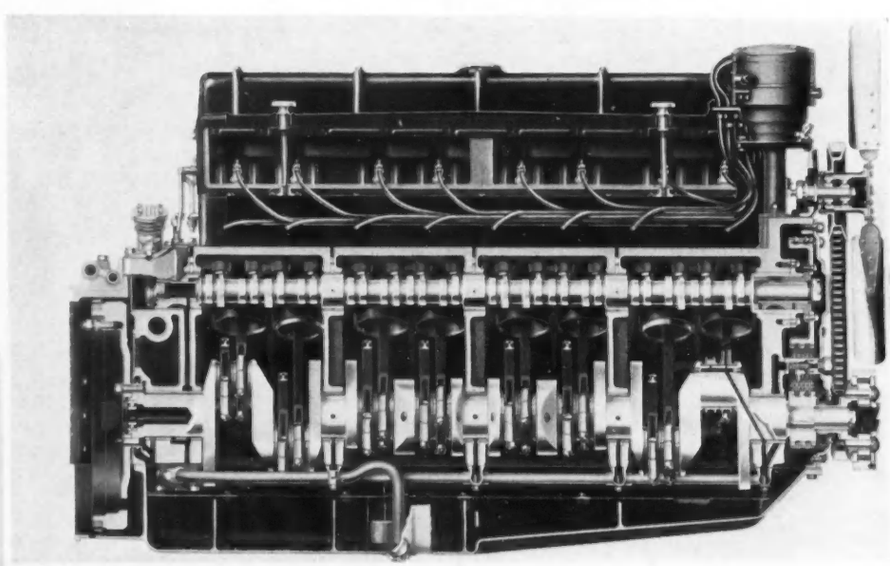

FIG. 5—VERTICAL LONGITUDINAL SECTION OF THE V-16 ENGINE

This Shows the Overlap of the Crankshaft Bearings with a 4-In. Piston-Stroke and, at the Front End, the Oil Channel through the Crankshaft and Connecting-Rod Leading to the Wristpin Bearing

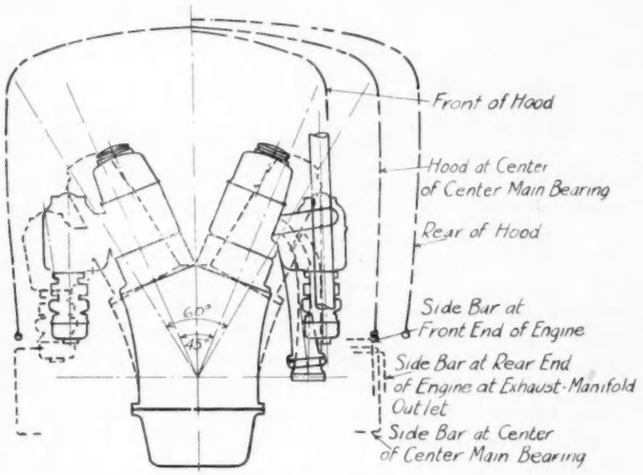
Front of Hood
Hood at Center of Center Main Bearing
Rear of Hood
Side Bar at Front End of Engine
Side Bar at Rear End of Engine at Exhaust-Manifold Outlet
Side Bar at Center of Center Main Bearing
60°
45°

FIG. 6—DIAGRAM OF V-16 45-DEG. ENGINE AND ITS ENVIRONMENT UNDER THE HOOD

The 45-Deg. Angle Gives Correct Clearances for the Frame, Hood and Steering-Gear, Whereas a 60-Deg. Engine, Indicated by Dotted Lines, Encroaches on All Three

head, (*c*) faster heat conduction from the piston to the cylinder-wall, and (*d*) cooler exhaust valves, gives a substantial cooling factor that allows a higher volumetric compression, or compression ratio, to be used. Therefore the compression ratio of the V-16 is 5.5:1.

With cooler pistons and lighter reciprocating parts, lower oil-temperatures are obtained.

Final Determination of Mechanical Layout

The layout or mounting of a V-type engine in the car and on the chassis is a very important preliminary to a decision as to the type of engine to be chosen from the list of acceptable designs. First, the low center of gravity of the V-type is desirable. For a given piston displacement the center of gravity is also lower with the multi-cylinder type, as the stroke-bore ratio is rather well standardized and the length of the stroke decreases with the diameter of the bore. This gives more oil space and lower cylinder-blocks or provides more ground clearance for an engine as the number of cylinders is increased. In the V-16 engine, with a connecting-rod-length ratio of 2.2 and a bore-stroke ratio of 0.75, the rods could be 2.2 in. shorter than in the V-8; actually they are 9¼ in. long.

The 4-in. stroke of a V-16 also was favorable to the design of the crankshaft, which, with a 2⅝-in. main bearing and a 2½-in. throw bearing, gives an overlap, without fillets, of 9/16 in. (See Figs. 4 and 5.) The compactness of the V-type engine gives the minimum of crankshaft length. With the overlap of the main and throw bearings and the reduced amount of counterweighting required, the torsional periods were very small and only required a small harmonic balancer in front, combined with the fan pulley, to eliminate the torsional period. The counterweights are built in-

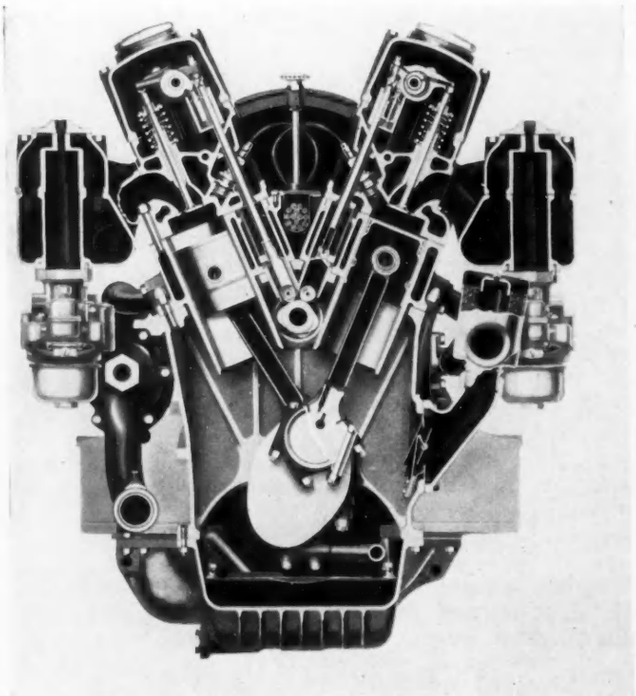

FIG. 7—CROSS-SECTION THROUGH VALVE MECHANISM, CYLINDERS AND MANIFOLDS

The Manifolds and Carbureters Are Placed on the Outside of the Cylinder Banks, Where They Are Very Accessible and the Carbureters Are Kept Relatively Cool. The Cylinder-Heads and Valve Mechanism also Are Readily Accessible

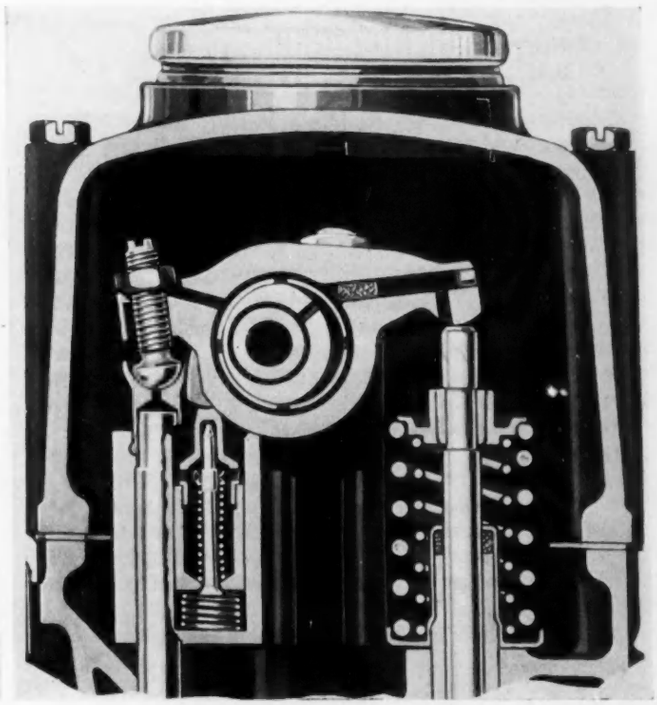

FIG. 8—DETAIL OF HYDRAULIC AND MECHANICAL VALVE-SILENCING DEVICE

The Eccentric Bushing of the Cross-Arm Is Rotated by a Light Spring-Pressure To Take up Clearance. A Hydraulic Plunger at the Lower Left Follows the Cam on the Bushing as the Relief-Valve Automatically Opens and Resists the Pressure Tending to Rotate the Bushing When Lifting the Valve

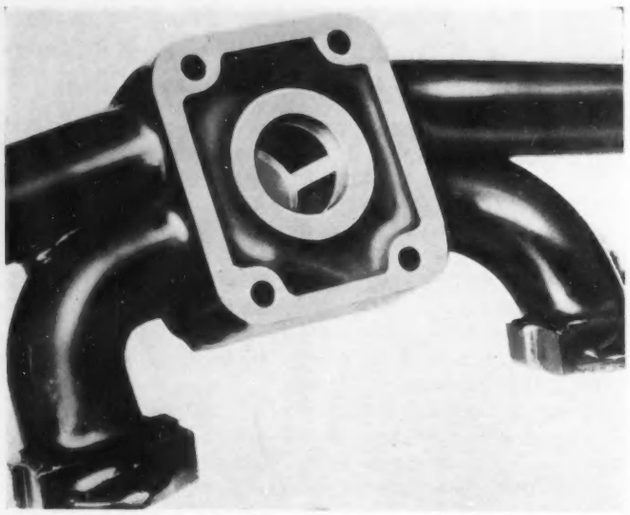

FIG. 9—DUAL INLET MANIFOLD THAT WAS FOUND TO GIVE BEST RESULTS

Both Manifolds Take the Mixture from the Central Chamber, from Which Two Openings Feed to the Forward and Two to the Rear Cylinders. The Branches of One Manifold Lead to the Center Four Cylinders and Those of the Other Manifold Lead to the Two End Pairs of Cylinders

tegral with the crankshaft cheeks, as shown in Fig. 4.

The controlling environment for the engine on the chassis and under the hood is shown in Fig. 6. The 45-deg. angle, indicated by solid lines, gives correct clearances for the frame, hood and steering-gear, while a 60-deg. engine, indicated by dotted lines, encroaches on all three.

Valves and Valve Mechanism

The 90-deg. V-8 engine has a V alley that sufficiently restricted the development of the carbureter, intake and exhaust manifolds and the valve mechanism; hence the consideration of any reduction of the available space meant a very close and not too satisfactory layout for even the 60-deg. angle. In the consideration of the 45-deg. angle, overhead valves were at once indicated and thought was given to combining the two banks of cylinders in one block having a 28-deg. V-angle. This idea was discarded, however, because of the anticipated difficulty in designing the manifold and its subsequent operation. This seemed likely to involve long research, with little hope of a successful solution.

Overhead valves were decided upon, first, because of their accessibility; secondly, on account of the desirability of placing the manifolding on the outside of the cylinder banks in two independent systems; and, thirdly, because of knowledge of the development of the automatic hydraulic valve-silencer by the General Motors Research Corp.

A cross-section of the valve mechanism in general is shown in Fig. 7, and the hydraulic and mechanical valve-silencer device is shown in detail in Fig. 8. The cross-arm is mounted on an eccentric bushing, which is rotated by a light spring-pressure to take up the clearance. The hydraulic plunger at the lower left center quickly follows the cam on the eccentric bushing in an upward direction as the small relief-valve automatically opens. In lifting the valve, the pressure tending to rotate the eccentric bushing in the opposite direction is resisted by the comparative solidity of the hydraulic plunger. Compensation for a condition that would cause riding of the valve is provided for in the fit of the hydraulic plunger, which allows sufficient leakage under pressure to give the necessary relief in a few strokes of the mechanism without being great enough to destroy its immobility. These limits are closely held, the parts are carefully made and the oil for this mechanism comes direct from the oil-filter.

Referring again to Fig. 6, the reader will see that the V-16 45-deg.-angle engine fits the chassis and the hood, allows a good layout in the V for the camshaft, valve rods, roller plungers, spark-plugs, wiring and so forth, and leaves space to attach the manifolding on the outside. The mounting of the accessories on the sides in the open spaces makes them and the cylinder-heads very accessible and presents a clean appearance. The engine is simple and, from an engineering viewpoint, is desirable for a number of reasons.

Following customary body practice, wood and clay models were made and installed in a chassis, and the external details were worked out with simplicity, serviceability and appearance uppermost in mind, so that when experimental models came through nothing had been left to chance. The appearance of the models completely confirmed the preliminary studies and "sold" the program for experimental production.

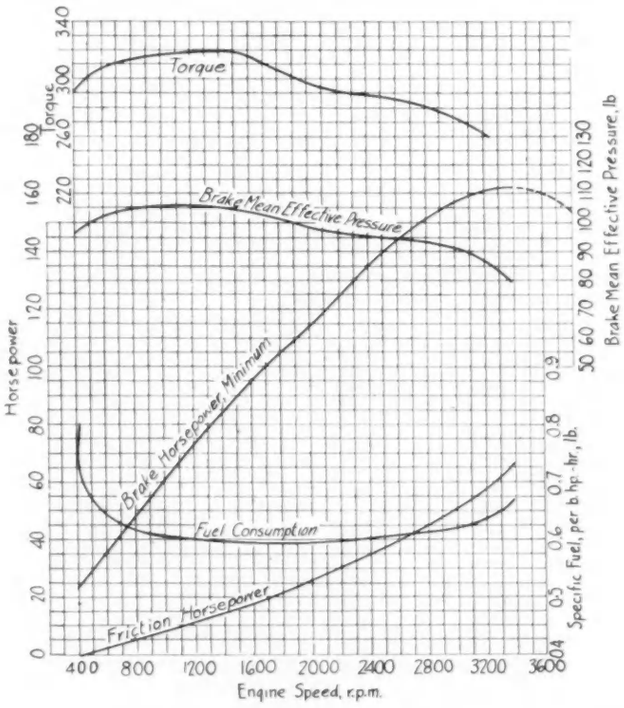
Torque
Brake Mean Effective Pressure
Brake Horsepower, Minimum
Fuel Consumption
Friction Horsepower
Horsepower
Torque
Brake Mean Effective Pressure, lb
Specific Fuel, per b hp.-hr., lb.
Engine Speed, r.p.m.
0 20 40 60 80 100 120 140 160 180 220 260 300 340
30 60 70 80 90 100 110 120 130
0.4 0.5 0.6 0.7 0.8 0.9
400 800 1200 1600 2000 2400 2800 3200 3600

FIG. 10—POWER CURVES OF THE ENGINE

Oil Control and Carburetion

One of the bugaboos of the multi-cylinder idea has, in the past, been control of the lubricating oil. To those who did not know how to control oil, the 8-cylinder engine was an impossibility, but the company with which I am connected and other companies overcame this with the eight and, in addition, made great progress in the methods and accuracy of making rings and cylin-

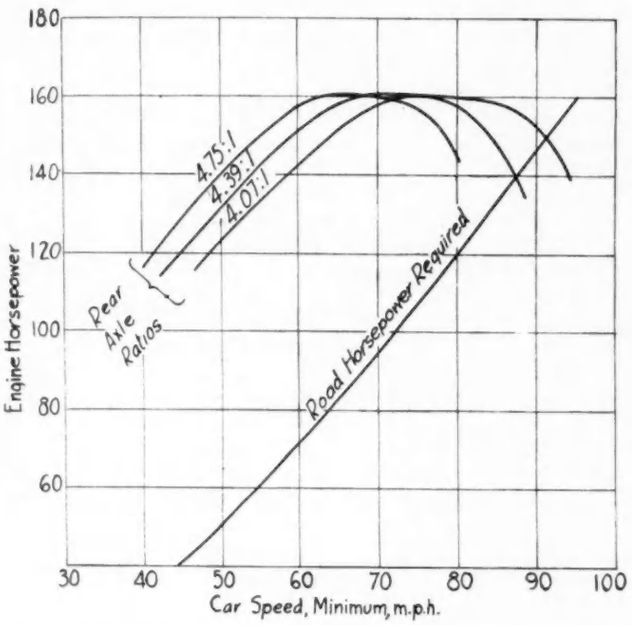

FIG. 11—SPEED POSSIBILITIES OF THE V-16 SEVEN-PASSENGER SEDAN WITH DIFFERENT REAR-AXLE GEAR-RATIOS

der bores. These later developments have assured the success of engines with even more cylinders and we present the V-16 with full confidence that oil control is an established fact.

Some of the multi-cylinder failures of the past, especially of the 12-cylinder engine, were due to incorrect design and inaccuracy in manufacture of the rings and bores, or to the fact that those parts did not hold their shape or necessary characteristics under power and heat conditions or after subjection to those conditions.

With the manifolds and carbureter on the outside of the engine next to the hood, we anticipated that special provision would have to be made to counteract the lower temperature of the air that entered when starting and warming up. The amount of the exhaust heat and the disposition of the exhaust manifold have not only compensated for this, but have given us an engine that starts easier under all circumstances, without any handicap on the development of more power with the cooler air available at full power and speed.

Several types of carbureter were tried, these being selected to suit the manifolds made ready for that purpose, but the Cadillac single-air-valve type, with a few modifications, proved to be the best, for we have, through several years of development, incorporated in its design many automatic and thermostatically controlled elements that give us the performance we want when starting either hot or cold, pick-up at any speed, and economy under full-speed conditions, which the air-valve type takes care of without spoiling low-speed performance.

Having two carbureters on this engine causes no more concern than a single carbureter, because we have so simplified and perfected the carbureter through its years of development that they will give us no more trouble than any repetition of a perfect article.

Synchronism is another bugaboo that has been buried. Idling is controlled by adjustable stops on each carbureter, and the throttle always comes back to the same place. Opening of the throttles equally from there on is secured by an initial adjustment that gives us as close control as is required.

Manifolding, Fuel System and Ignition

With a bank of eight cylinders on each side, symmetrical manifolding was easily designed for each system of eight cylinders. The type found to give the best results is virtually a dual manifold as shown in Fig. 9, one for the center four cylinders and one for the two end pairs of cylinders. Both manifolds get the mixture at a central chamber directly over the single carbureter, there being two openings feeding to the forward and two feeding to the rear cylinders. In other words, the single carbureter, with an exhaust-heated riser of sufficient length, feeds through four symmetrical openings into the four ways of the manifold, each leg feeding two cylinders through a Siamesed pair of inlet-valve chambers.

Two vacuum tanks are mounted on the dash, one for each carbureter. Sufficient gasoline flow for wide-open-throttle operation is provided by the auxiliary vacuum pump, which is of sufficient capacity for continuous full-speed performance. With the carbureter, vacuum tanks and leads disposed on the outside of the engine close to the hood, the gasoline is never overheated, but provision is made for the escape of any gas that might form.

An ignition distributor-head of larger diameter than on the V-8 engine was used, and a large eight-lobed cam operating two breaker-points gives correct ignition up to new high engine-speeds. Our calculations were based on 4000 r.p.m., but this speed has been considerably exceeded.

Results Obtained with 16 Cylinders

The smoothness of operation obtained gives the term a new significance; silkiness would be a more appropriate word. The pick-up, the running on the road and the over-running are all smooth and quiet.

Our expectations as to compression were realized, the ratio of 5.5:1 proving satisfactory, especially for an overhead-valve engine. A power curve of the V-16 model 452 is reproduced in Fig. 10, and Fig. 11 shows the speed possibilities of the car with different rear-axle gear-ratios.

Our standard gear-ratio gives very good all-round results, the acceleration with the seven-passenger sedan body being 4 ft. per sec. per sec. or better, with a top speed of well over 80 m.p.h. Tests of over 90 m.p.h. have been made with standard bodies, and 100 m.p.h. with roadsters.

The fuel economy at medium car-speeds has equalled that of the V-8, but as a rule, especially at highest speeds, the mileage is slightly less; however, this is only a nominal price to pay for the increased comfort and speed of performance.

THE DISCUSSION

QUESTION:—With the spark-plugs and wires covered up in the V between the cylinder banks, how is cooling of the plugs accomplished?

W. R. STRICKLAND:—They never were cooled by air; the water-cooling takes care of the plugs.

QUESTION:—Do the valves normally operate with no

clearance between the rocker-arm and the valve stem?

MR. STRICKLAND:—There is virtually no clearance.

QUESTION:—Will you tell something more about the material and design of the pistons, and how the lubrication problem was solved?

MR. STRICKLAND:—Having worked with cast-iron pistons, we feel that our problems are considerably lessened. The nickel iron which we have used for many years, together with the design of the piston, enables the piston to hold its shape under heat conditions. That has been studied very carefully, because holding the shape under heat conditions is one of the biggest problems we have with any piston. When these pistons are exposed to the extreme heat, they do not bear against the cylinder bore in any particular line or spot; they show even bearing surfaces, and therefore the clearances can be normally lower.

That in itself has a great deal to do with controlling the oil, but other factors are the design of the bores; keeping them round in the manufacture; holding them so that they do not go out of shape when the cylinders are bolted down or under the heat of operation, as a great many cylinders do; improvements in piston-ring design; and careful selection of rings to properly fit the bores, which entails a great deal of work in the inspection departments. Careful fitting of the rings is a necessary part, as is also control of the oil thrown off by the crankshaft or connecting-rods.

We have plenty of oil; we lead it up to the wristpin through holes in the connecting-rods. We also have what we call a small bleeder-hole; sometimes we need it and sometimes we do not; we have it on the eight-cylinder engine. The heat conditions require plenty of oil, but it must be controlled. We have succeeded in controlling oil to such an extent that doubling the number of cylinders does not worry us; we do not hesitate to take the step, and have not had any difficulty since doing so. We operate the 16-cylinder cars on the proving ground, day and night, for thousands of miles, and we know whereof we speak in that regard.

QUESTION:—Have you had any trouble with carbon formation filling up the slots in the oil rings?

MR. STRICKLAND:—We have not had any trouble with that. It is conceivable that we may have some in the course of time, say after the car has been run in the neighborhood of 25,000 miles, which we would consider sufficient for any engine before overhauling. (Since the meeting in April it has been learned that engines that have operated up to 50,000 miles are still free from such trouble.—W. R. S.)

Geared Centrifugal Superchargers

(*Concluded from p.* 153)

of compression. Approximately, the friction drop would vary with the square of the rotative speed. The pressure rise produced by a geared supercharger also varies approximately with the square of the rotative speed. Hence, at every speed, the supercharger pressure-rise will exactly balance the friction drop and there will be atmospheric pressure in the cylinders at all speeds. Such a supercharger will give exactly constant torque and power varying directly with the speed, up to the limit at which it is safe to operate the engine. There would be no tendency toward detonation in such an engine, as the compression pressure would be constant at all speeds. Some of the supercharged engines now in commercial use, having supercharger gear-ratios of 7 to 1 and 8 to 1, closely approximate this desirable condition.

If a somewhat greater amount of supercharging is provided than that already assumed, then, as the engine speed is increased, the supercharger pressure will more than counterbalance the inlet friction, so that the cylinder pressure at the beginning of compression will successively increase with the speed. A limited amount of such excess supercharging will not cause detonation, as the compression pressure at which detonation occurs increases with the speed. Such an amount of supercharging will give increase of torque with speed increase. A limited amount of this will cause no difficulty, but, if further amounts of supercharging are attempted, compression will increase to such an extent at the higher speeds as to cause detonation. If such an amount of supercharging is necessary, antiknock compound must be added to the fuel.

No doubt a number of satisfactory airplane engines without superchargers are existent, but the number of engines using superchargers and the rate at which this number is increasing constitute proof that superchargers give an important advantage in connection with the four items first mentioned. Any good engine not equipped with a supercharger will be a better engine if a supercharger is installed.

Constant Mesh *or* Sliding-Gear Transmissions

By F. C. PEARSON[1]

Semi-Annual Meeting Paper

Illustrated with Photograph

ONE OF THE most important developments in the passenger-car field in the last year has been the production of better transmissions. When the three-speed sliding gears superseded the two-speed planetary transmission, the development was signaled as an engineering achievement. Several years later the four-speed transmission was given much publicity, but with the advent of more cylinders and better engines, transmissions returned until recently to the conventional three-speed type. We are now in another era of transmission change. Developments point this time toward better, quieter and more useful transmissions, whether they are built with three or four forward speeds.

Road conditions have improved to such an extent in the last few years that cars are now constantly being driven at higher average speeds than formerly. This high speed greatly lessens the life of engines, making necessary either the building of larger engines that will prove more expensive to operate or the gearing of cars so that high speeds can be obtained without too high engine-speed. The present tendency is to build transmissions that will have high-gear quietness in second or third gear, which makes possible the use of lower ratios when desirable.

Until recently all of the well-known cars used spur-gear clash-type transmissions. The undesirable characteristics of this transmission need not be mentioned, as we all use this type. Engineers have recognized that if we could build a transmission that would be free from the shifting difficulties and the noise while running in gear, a great advance would be made. Apparently such advancement has been made in at least three types, known as the internal, herringbone and helical gear, which are now in production. All these overcome, to a great extent, the undesirable characteristics mentioned. Noise is overcome by the use of special types of gear. They are all of the constant-mesh type and shifting is done through multi-jaw clutches that shift at virtually any speed and require little or no practice by the operator to enable him to make quick, smooth gear-changes.

Since no clash-type transmission that gives the results mentioned has been developed, constant-mesh clutch-shifted transmissions may be considered to have advantages that cannot be incorporated in the clash type.

The constant-mesh type of gearing and shifting has been used only for high and second speeds. Practice seems to prove that carrying this through the first and reverse gears is neither necessary nor economical. As shifting these gears is almost always done while the car is standing still, their engagement offers no difficulties.

After considerable research and study, the company with which I am connected started to develop the herringbone type of transmission, the advantages of which over the internal-gear type are simplicity, lower cost and less weight. As the same number of gears and bearings are used as in the conventional clash-type, the bugbear of complicated construction did not exist.

Gear thrusts are always present when helical gears are used, but are not present in the herringbone construction, since each pair of gears absorbs the driving thrust within itself. This is an important point, as balancing the end thrust arising when driving through two pairs of plain helical gears is impossible and thrust bearings have to be relied upon to take the unbalanced thrust-load and maintain the location of the gears.

The multi-jaw clutches used in the three constant-mesh types of transmission seem to be very similar in design except for jaw shapes, which have been worked out to fit their individual cases.

Many combinations of tooth forms and pitches were tried during the development work on the herringbone transmission. Experiments were made with gears varying from 8 to 20 diametral pitch. Results show that, in the case of plain-type herringbone gears, the finer the pitch is, the quieter are the gears. This fact really hinges upon the width of tooth required to transmit a given load. A coarse pitch permits the use of a narrower gear than a fine pitch designed to carry the same load. This results in a greater tooth-overlap with the fine pitch. With the high tooth-angles used on these gears, the tooth overlap seems to be an important factor in the vibration period of the gear train as a unit. However, as the pitch is made finer, the strength and particularly the fatigue resistance diminishes.

The next problem was to determine how small a pitch can safely be used to carry the required power. When this question had been settled, the gears were not as quiet as we had hoped, although they were far better than those of any conventional transmission.

Two Pitches for Second-Speed Gears

Our next line of research was a refinement of the first design. We found, after many experiments, that using two pitches in both constant-mesh and second-speed gears was desirable. However, the greatest improvement is shown when two pitches are used in the second-speed test. We also found that the selection of the odd pitch in the second-speed gear-train was limited, so a final decision depended upon the well-known cut-and-try process rather than on mathematical calculation. A study of periods of vibration of the individual parts gave no satisfactory results. We know that the periods of vibration of the gears and the case have an all-important relation; even spur gears are quiet while transmitting power if they are mounted in an abnormally heavy case. This shows that the problem is one

[1] Assistant chief engineer, passenger car division, Reo Motor Car Co., Lansing, Mich.

of reducing the effect of vibration of one part upon another.

The real key to such success as we have attained in eliminating gear noise depends entirely upon the dampening of these disturbing vibrations by the odd-pitch combination. The theory is that the transmission of power by gearing is a step-by-step process in which each tooth is a step. As all gears are to some extent inaccurate, we have a series of blows spaced the same distance apart which set up a vibration in the case. An example of this is an army of men marching in step over a bridge; what this rhythmic vibration can do to structures of this sort is well known. By using two pitches of gear teeth we produce a result that is similar to that of the men on the bridge when they break step.

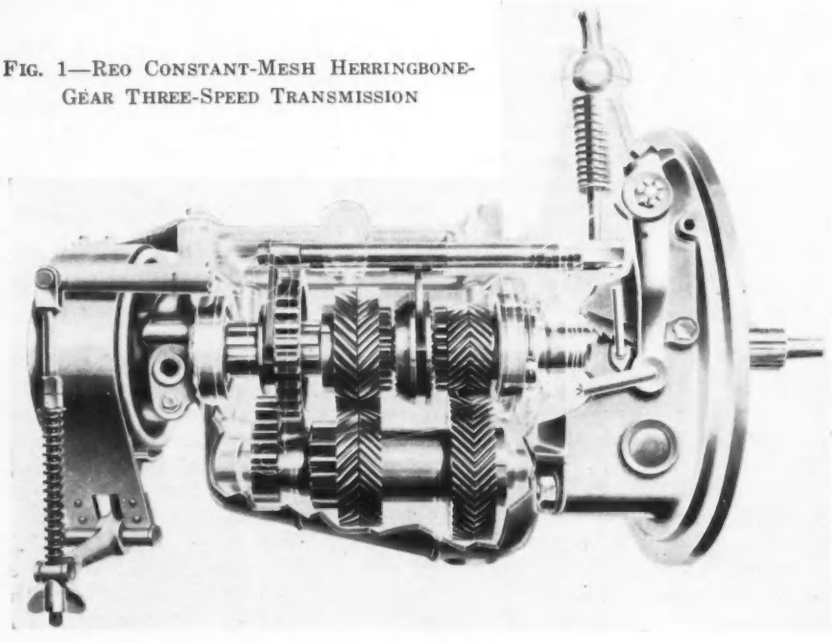

FIG. 1—REO CONSTANT-MESH HERRINGBONE-GEAR THREE-SPEED TRANSMISSION

Construction of the Herringbone Train

Briefly, the construction of the herringbone train consists of two pairs of gears: the constant-mesh and the second-speed. No other description is needed, as the low and reverse-speed pairs are conventional spur-type gears. The main-drive gear is the only one that is fixed in location, the front ball-bearing holding this in place. The two gears on the countershaft are fixed in relation to each other, but the countershaft can float endwise. The second-speed gear floats on the main spline-shaft, as far as endwise motion is concerned, and power is transmitted from it to the driveshaft through a dog clutch.

Each gear is made of two helical gears of opposite angle placed side by side. No difficulties of locating them occur, since each gear centers itself because of the V-shaped angle of the teeth. Each pair of half-gears is keyed to its respective shafts, and the keyway is simultaneously broached in a pair of half-gears in a machine that locates the teeth with respect to each other. Each of the four gear-units is treated in the same way. The next operation is to roll the constant-mesh or the second-speed pairs together to select them for center distance, clearance and tooth contact. After this operation, the pairs are never separated while passing through heat-treating, lapping and assembling operations. This seems to be the only way to maintain the uniformity of production that is necessary to keep noise at the minimum so that the percentage of tear-downs will be low.

Assembling, Testing, and Material Costs

Assembling is no more expensive or difficult than for the conventional transmission. Testing still presents the same difficulties regarding noise as with the spur-gear transmission. It is done by ear in a quiet room at the end of the transmission-assembly line. While the operator is unconscious of weather changes, we have not as yet overcome the effects of indigestion, so the usual arguments arise now and then regarding the occasional passing or rejecting of transmissions that border on the maximum-noise limit.

The bearing loading is slightly greater for herringbone than for spur gears in transmissions in which the center distances between the main shaft and the countershaft are the same. Transmissions of this type are driven in gear probably 50 per cent more and much faster than is the conventional type; consequently, more bearings are required to maintain the same life.

A word about materials may not be amiss. Using any other than the standard steels regularly used by conventional transmission-makers has not been found necessary. The design of this transmission allows us to use bar stock and make all of the gear blanks in automatic machines. This results in a saving over the cost of forged blanks and seems to give us gears that are more uniform in grain structure and have less eccentricity after heat-treatment.

THE DISCUSSION

L. OCHTMAN, JR.[2]:—The illustration shows that the two halves of one pair of the herringbone gears have different tooth-pitches, while in the other pair the teeth apparently have the same pitch. Will Mr. Pearson explain that?

F. C. PEARSON:—Different pitches are used only on the second-speed set. Anyone who is well acquainted with gear-cutting knows that to cut gears that are not a 1:1 ratio would require a very peculiar set of cutters to cut two different pitches and mate them together. When we first brought out the herringbone transmission we had no cutters of that type, but today we have such cutters and can cut two different pitches on pairs of gears that are not a 1:1 ratio. I think you can easily see why, if the ratio is 1:1, the teeth can be cut with the standard type of cutter. The 1:1 ratio is the only one that can be cut with the standard cutter; any other ratio would require special cutters.

[2] M.S.A.E.—Chief engineer, Elco Works of the Electric Boat Co., Plainfield, N. J.

American Passenger-Car Gearsets

Discussion on Herbert Chase's Semi-Annual Meeting Paper[1]

WRITTEN discussion of the paper deals with the advantages of internal-geared transmissions, the superiority of helical and herringbone gears over straight spur-gears, a gearless transmission used in conjunction with one-pedal control, the lower cost of producing an acceptable, quiet internal-geared transmission than of producing other types, and the importance of comparatively small axle-reduction and quiet next-to-direct gearing for both three-speed and four-speed transmissions.

In the oral discussion a description is given and comments made on an English "fluid flywheel" or slipping clutch used in conjunction with a transmission, arguments are advanced for and against providing more power to secure the "punch" that drivers want without adding complication and requiring more gear-shifting, a two-speed device on the transmission to give two different driving-speed ranges is advocated, and the four-speed transmission is asserted to be simpler than the three-speed. Lowering the engine speed through the use of a four-speed transmission and a faster rear-axle ratio is stated to increase the stresses through the driving train. With cars of a larger power-weight ratio, the need of four speeds to get plenty of "punch" at high speed is said not to be apparent.

What the public wants, rather than the technical phases, will decide the question, in the opinion of several speakers. It is conceded that car owners want quick acceleration and good hill-climbing ability, but do not want to have to shift gears any more frequently than they are accustomed to doing.

THE WRITTEN DISCUSSION

S. O. WHITE[2]:—Until within the last few years passenger-car transmission design followed a few rather well-beaten paths and consisted largely in trying to get the most capacity possible into the least space, with the lightest gears and most compact and economical arrangement consistent with reasonable service performance. Constant progress in the manufacture and heat-treating of gear steels and in the gear-cutting art accompanied this development in design. Three forward speeds and a certain amount of gear noise were generally accepted in this Country, although with the feeling that a way might be found to reduce the noise.

S. O. WHITE

Then the quiet-running properties of internal gearing were called to our attention, and a revival of the four-speed transmission followed, but with the internal gear used for the next-to-top speed. Four-speed transmissions for passenger-cars were not new and their advantages were known, but their noisiness stood in the way of their acceptance. The provision of a sure and easy shift between the two top speeds was essential, and fortunately the nature of the internal-gear design made this fairly easy. As a consequence, several four-speed internal-geared transmissions have come on the market and have achieved noteworthy popularity. To get the greatest benefit of such a design, the axle ratio should be higher than was common practice.

Mr. Chase has set forth very concisely eight advantages resulting from such a combination. I doubt very much whether these advantages are generally known, as they should have led to even greater popularity for the four-speed idea. Possibly some of the publicity has been too technical and, as the public gains experience and becomes four-speed wise, the demand may be greater. Evidently the disadvantage of sluggishness in high speed, resulting from the high axle-ratio, and the consequent increase in the amount of gear-changing necessary, have militated strongly against this type of transmission.

However, as Mr. Chase brought out in his paper, the advocates of three-speed transmissions in combination with fast rear-axle ratios secure the desired performance in high gear by increasing the ratio between the car weight and the engine capacity. Nothing prevents doing the same thing for a four-speed car, thus retaining high-speed ability and also the various advantages that go with the four-speed fast-axle combination. Different car characteristics call for various combinations, and the same formula cannot be applied to all cars with the same results.

Even after thorough trial of the four-speed transmission, many engineers are unconvinced and probably always will be. On the other hand, some who were formerly opposed are now converted after having had more experience. To a certain extent our organization has found the situation to be a matter of personal engineering opinion and of the kind of driving and performance each one prefers. Some approve the four-speed principle but hold that the cost of the internal-gear construction is prohibitive; they would use some other form of gearing and tolerate slightly more noise.

The success of internal gearing has stimulated activity with special designs of spur, helical and herringbone gears for application to either three-speed or four-

[1] The paper was published in the S.A.E. JOURNAL, June, 1930, p. 727. The author is associated editor of *Product Engineering* and *American Machinist* and is a Member of the Society. The main points in the accompanying discussion are indicated in the abstract printed above. Part of the discussion was submitted as prepared discussion and part was given extemporaneously at the meeting.

[2] M.S.A.E.—Chief engineer, Warner Gear Co., Muncie. Ind.

speed transmissions. As a result of these efforts, transmissions of the various types are in production. Use of helical and herringbone gears usually is confined to the two top speeds, the gears being in constant mesh. Various forms of dental or jaw clutches are available to make an easier shift between the two top speeds.

Helical and Herringbone Gears

In applying either helical or herringbone gears to a transmission, very careful consideration must be given to relative manufacturing difficulties. Each presents its peculiar problems, which are not quite as simple as those of straight spur gears, but they have been found to be so much better than spur gears as regards sound, at the same time being cheaper than internal-gear constructions, that they seem certain to increase in popularity for both three-speed and four-speed transmissions.

Experience with helical-gear designs has not indicated any practical difficulty from end thrust. On the countershaft cluster the thrusts of the two helical gears are opposed and virtually balanced. On the main shaft the thrusts are taken directly on annular ball-bearings in conventional mountings, and the size of these bearings must be increased to provide ample thrust capacity.

For many years the car manufacturers thought that they could not afford to put much added cost into transmission specialties. The gearshift was the favorite point of attack for the prolific inventor, and many kinds of mechanical gearshifts were offered to our company. Some of those worked very well but none made any substantial progress. It remained for the internal-geared four-speed transmission to break the ice and prove that the public desire is for something better than the conventional transmission.

To sum up the case for and against three speeds and four speeds, something is to be said for each. Unquestionably, slowing down the engine makes a sweeter-running automobile. Again, multiple speeds, properly used in individual chassis designs, can improve car performance. Engineering opinion has held that the public does not want to shift gears. With the advent of easier gearshifting, this opinion should perhaps be modified, as the public will render the final decision.

Gearless Transmission and One-Pedal Control

A. BJORKMAN[2]:—Rapid acceleration no doubt is important, especially in crowded city traffic, where frequent starts have to be made and ready adaptability to the traffic changes is highly desirable. As the speeding-up of a car represents mechanical work performed by an engine of limited capacity, it is important for rapid acceleration that the engine should be run as close to its maximum output as possible during the whole acceleration period so as to deliver the necessary work in the shortest possible time. This calls for a continuously varying gear-ratio. A four-speed transmission gives the driver possibilities of keeping the gear-ratio closer to the ideal than does the three-speed gearset, but, as Mr. Chase points out, whether the greater possibilities of the four-speed transmission in this respect are utilized by frequent gearshifting depends on the driver.

Many drivers take delight in gearshifting, displaying a high degree of skill in choosing the right moment to shift, but in spite of this fact I think the motoring public would gladly take to a transmission that would altogether relieve the driver of shifting gears, especially if such a transmission could increase the rate of acceleration by maintaining the ideal gear-ratio during the acceleration period. Several attempts have been made to design such a device, the latest and perhaps most promising being the Spontan transmission, which has neither clutch nor gears, its operation depending on the centrifugal forces of two rotating weights acting on two overrunning roller clutches. The use of centrifugal forces for transmitting the drive makes the action extremely smooth and flexible at all speeds. Because of the absence of gears, it is noiseless, and its efficiency is about 98 per cent.

From the acceleration point of view, this transmission is especially interesting. With the car standing still and the engine idling, the centrifugal forces are negligible and the engine is, in effect, free from the driveshaft. This fact enables it rapidly to gain speed when the accelerator pedal is depressed, thus creating high centrifugal forces and a powerful starting torque, giving a quick but smooth start. The amount of the starting torque may be judged by the fact that an Essex Super Six four-door sedan with standard rear-axle gear-ratio, fitted with this transmission, was repeatedly started and accelerated on a 46-per cent incline without difficulty. At every instant during the acceleration period the transmission is automatically supplying the best gear-ratio for that moment, resulting in a remarkably good pick-up. This applies to both forward and backward drive.

During periods when less driving torque is required, the transmission changes to direct drive, cutting out the action of the main working parts. This is an important feature, as it enables those vital elements, the roller clutches, to be designed to last the whole lifetime of a car.

Important as the technical advantages of increased acceleration and smoothness no doubt are, the main advantage that this transmission offers is the greatly simplified handling of the car as a result of abolishing gearshifting. The transmission is foolproof and prevents the engine from being stalled. Moreover, the inherent features of coasting when the engine is not driving and an automatic no-back device prove very useful, especially in hilly country.

With a view to simplifying car control as far as possible, the Swedish company has worked out a one-pedal control for motor-cars embodying its transmission. This design reduces all of the driving operations—starting, accelerating, speed changing, stopping, braking and reversing—to the simple manipulation of one pedal. Pressing down the pedal causes the car to accelerate, whereas the negative action of releasing the pedal, by hesitation or otherwise, causes the car to slow down. The one-pedal control is used in the same way as the accelerator pedal of ordinary cars, the action differing in that an extended release of the accelerator pedal increases the deceleration by setting spring-operated brakes in action, thus saving the driver the time and trouble of moving his foot over to the brake pedal. Incidentally, the one-pedal system has, besides trial in a number of experimental cars, been in successful operation in a taxicab in Stockholm, Sweden, for one year, covering about 30,000 miles in taxicab service, and the drivers stated that they prefer it to the ordinary type of control.

[2] Research engineer, Spontan Co., Stockholm, Sweden.

Reasons for Preferring Internal Type

C. A. NERACHER[4]:—We have four general types of transmission from which to make selection. These are the spur, herringbone, helical and internal-gear types. In selecting the most suitable one, the major considerations are general excellence balanced against cost. In general, the cheapest is the conventional spur-gear type and the most expensive is the internal-gear type, with the herringbone and helical-gear types coming in between. These cost comparisons are based only on design. To produce a very quiet transmission makes the cost a very different consideration, in which a more expensive design may be the cheaper to produce.

The company with which I am connected selected a more expensive type from the standpoint of design. The difference in ultimate cost, if any, must be justified by greater excellence, expressed in terms of public appeal and sales advantage. The corporation has been the largest producer of the internal-gear type in the last 18 months, and as a result of this experience is adding to its line one smaller size of transmission of this type. The reason for continuing this type in the face of the general opinion which is based only on design comparisons, that the cost in production is greater than that of other types, is that our experience indicates that ultimately it will be cheaper to produce an acceptably quiet job of this type with commercial limits than it will be to produce other types. By commercial limits I mean the limits that can be maintained with normal scrappage and favorable cost in a well-equipped and maintained gear-shop.

C. A. NERACHER

The great mass of buyers who make this industry possible are fairly conservative, and general driving and riding comfort are more important to them than the last ounce of performance; therefore I believe that the engineers are too uncompromising when balancing performances against comfort. My particular reference is to engine speed and gear ratios. I believe that a car rapidly accelerated up to 50 m.p.h. on a good third-speed and then at a slower rate of acceleration through fourth speed is more desirable to the large majority of the buying public than a combination giving maximum performance at the upper end of the range with high engine-speeds.

The character of the third speed required to meet trade demands and promote sales must be so good that the car operator can stay on it, when conditions require that speed, with no annoyance or nervous fatigue. In other words, it must be about as good as the direct drive. I believe the possibilities of getting this result in production favor the internal-gear type and that next year's experience with transmissions of the different types will probably show that the more expensive design of internal-gear transmission may be the cheaper manufacturing undertaking when all things are considered.

The last 18 months has been a period of manufacturing development. Out of our own experience we feel we have gained the knowledge required to produce transmissions of this type covering the requirements herein mentioned. The public is rapidly being weaned away from axles that give high engine-speeds at high car-speeds, and the four-speed transmission has accomplished much of this.

Purpose of Cadillac Dashpots

C. V. CROCKETT[5]:—One rather small point to which I should like to call the author's attention is a not quite correct statement in his description of the Cadillac transmission. This reads: "The yoke mechanism includes a pair of oil plungers or dashpots arranged to automatically prevent excessively rapid shifting."

The dashpots are not of sufficient power to prevent rapid shifting in case an owner desires to do so. They are, however, the connecting link between the shifting mechanism and the friction-clutch control mechanism. The resistance of the dashpot determines the power with which the friction clutches will be engaged. While this is seemingly a trivial point, the entire success of the transmission depends upon these dashpots. If gears are shifted slowly, little force is needed, and, since the dashpot plunger will be depressed slowly, little force is applied. Upon making a rapid shift, however, the dashpot presses the clutch into much harder engagement, applying a more powerful synchronizing force which is needed under these conditions. In a like manner, resistance of cold oil in the plungers assures the added pressure necessary to synchronize gears rotating in cold oil, and hence having a tendency to stop quickly.

Small Axle-Reduction Urged

CARL E. SWENSON[6]:—I strongly urge the use of comparatively small axle-reductions as the best means of obtaining pleasant high-speed car operation. This naturally calls for a high power-weight ratio and, coupled with a gearset having a quiet next-to-high or direct gear-ratio, makes a car that is much more satisfactory to drive than one having conventional axle-ratios.

I drive cars of both types, and there is no question in my mind as to which gives the greater satisfaction. If, after driving the fast-axle car, I immediately drive the low-gear car, I at once get the feeling that the engine is racing. Also, the maker of the fast-axle car seems to have succeeded in providing an engine that makes the power-weight ratio such that the amount of gear changing is not noticeably greater than that to which I have always been accustomed.

Mr. Chase mentions in his paper that few persons drive at speeds higher than 50 m.p.h.; but at this speed, and at even lower speeds, it is much more enjoyable to drive a car having a well-balanced, comparatively slow-running engine. Although I do not believe that the average driver will be satisfied to drive more than is absolutely necessary in an indirect gear, I believe that it is but a matter of time before all of the better cars will have quiet gearing for the next-to-direct gear-ratio, for three-speed as well as four-speed gearsets.

F. H. JONES[7]:—There is no question that the pas-

[4] M.S.A.E.—Engineering division, Chrysler Motors, Highland Park, Mich.

[5] Jun. S.A.E. — Secretary, engineering department, Cadillac Motor Car Co., Detroit.

[6] M.S.A.E.—Chief engineer, Mechanics Universal Joint Co., Rockford, Ill.

[7] President, Jones Clutch & Gear, Inc., Upper Sandusky, Ohio.

senger-car transmission is somewhat "in the air," as so many different persons have their personal opinions and ideas regarding what they think should be correct and proper.

We have spent considerable time on the development of our three-speed and four-speed transmissions and believe that we have worked out a product that is very difficult to improve upon.

All our herringbone gears are made in two separate pieces, and we have eliminated any gap between the teeth so that, when rotating, the addenda of the teeth leave the mating gear on a film of oil. The loose or slipover gear is driven by four hardened dowels and is held in position by hardened nuts screwed up tight and securely locked.

We employ three different angles on our herringbone helix, as well as three different pitches on the three sets of gear trains. The finest pitch on each train is 13, 12 and 11 respectively. Our low-speed sliding and reverse gears are made 8 pitch. All gears have full-depth teeth, no stub teeth being used in these transmissions. Mr. Chase has given a good idea of the construction of this transmission in the supplement to his paper.

As to whether the drive should be direct in third or in fourth speed is, in most cases, a matter of opinion; some may desire the fourth speed to be direct and others that it be over-drive. This, of course, is determined mostly by whether the third speed is quiet when the transmission is designed for direct drive in fourth speed. In our over-drive, any gears in which we drive from 25 m.p.h. up is noiseless; and we have the same results on the third speed when drive is direct on fourth speed.

Our three-speed transmission has a noiseless second speed and we have calculated our ratios so that either third and fourth speed in the four-speed transmission or second and third in the three-speed transmission is adapted to continuous normal driving. We have made the second speed in our four-speed transmission such that it is possible to get a driving speed especially suitable for steep grades on which it is necessary to go into second. We have also found it practical to step up the ratios in the transmission, making the low-speed ratio in our four-speed transmission in which direct drive is on third about 2.28 to 2.35:1, and in some cases we make it as fast as 2.18:1.

With present road conditions, size of wheels, gear ratios and high-speed engines, it is not necessary to have a very low first speed. This gives the car a very fast getaway in low, and we use first, second and third speeds the same as one would with the conventional transmission. For speed on the straightaway, the transmission is shifted into the fourth position, which is an over-drive and makes a beautifully performing car, as first, second and third speeds should be used as usual for city driving and in traffic, eliminating gear-shifting as much as possible.

Some criticism was made in the paper that drivers do not like to shift gears. That is true with the conventional transmission if the clutch performance is bad, as it is difficult to make these shifts unless one is an expert, but with our transmissions no effort is required in shifting at any speed, and the shift from third to fourth speed or vice versa is often made unconsciously, especially if one is driving in the over-drive and comes to a steep grade that causes the car to slow up slightly. Fear of being unable to get into a lower gear is absolutely forgotten with this transmission.

Mr. Chase commented on the compactness of this transmission; however, it is very simple to take down and reassemble, being more nearly like the conventional transmission than are other four-speed gearsets. The overhang is not much greater than with the conventional three-speed transmission.

We are arranging to produce a four-speed transmission with direct drive in third and over-drive in fourth speed, that will replace, in the same space, the transmission now being used in Buick cars.

We like to aid in the progress being made to improve the comforts of driving by furnishing a transmission that has no irritating gear sounds when running or clashing of gears when gear changes are made. We are spending much money in the development of these transmissions, which have been tested for more than three years without discovering a weak point.

THE ORAL DISCUSSION

CHAIRMAN T. J. LITLE, JR.[8]:—Every engineer has had various types of automatic transmission submitted to him from time to time. For 10 years back great sums of money have been expended on the design and manufacture of some type of automatic transmission, yet we see nothing of note in production in that line. The most interesting device of that nature that I have heard of recently is the Daimler fluid flywheel which is described in the latest issue of *The Autocar*. I think it will receive considerable attention in American engineering circles very shortly. The fluid flywheel, so-called, is a very simple device that is used in conjunction with a transmission and utilizes the slip between two plates having cavities and using oil as an intervening medium for driving. These plates slip below a rate of 600 r.p.m. and gradually pick up to full drive at the higher speeds. This so-called fluid flywheel has given very fascinating driving results, I am told, and I rather expect that it will be used in this Country.

T. J. LITLE, JR.

I recommend that this article be reprinted or abstracted by some of our technical journals in this Country for what it is worth. I will read a section or two:

> To anyone who has spent years in the art of letting in clutches gently, and accelerating engines suitably, in order to move a loaded car away from rest with perfect smoothness, the Daimler fluid flywheel seems pleasantly uncanny.
>
> Imagine yourself to be in the driving seat of a luxurious big car. With the gear lever you engage

[8] M.S.A.E.—Engineer and industrialist, Holley Carburetor Co., Detroit.

top gear, then you switch on and start the engine. The clutch pedal you neglect entirely. When you want to start, you take off the hand brake, and then depress the accelerator pedal, gently or quickly, a little way or all the way; it does not matter, the result is the same; the car just glides away. It picks up speed with perfect smoothness — no shuddering, no hesitation. It floats quietly along on top gear. To go faster, depress the accelerator more; to reach maximum speed, press the pedal right down. To stop, release the accelerator and apply the brakes until the road wheels come to rest. To restart, release the brake and apply the accelerator.

W. R. STRICKLAND

The fluid flywheel transmits power in the following manner. Assuming the car to be stationary and the engine started, the rotation of the driving member by the engine causes the oil in its cells or cups to flow toward their outside periphery. From here, as the driven member is yet stationary, the oil flows past the outside periphery of its cups, through them, and past their inside periphery to the inside periphery of the driving-member cups, and from there back again to their outside periphery. In other words, the oil starts on a circulatory motion between the cups of the driving and driven members of the clutch.

In passing from the webs of the driving to those of the driven member, the oil is retarded in velocity, and therefore releases kinetic energy, which sets the driven member in motion. Since, even when the driven member has attained full speed, the load on it causes it to lag behind the driving member, the centrifugal forces in the latter are always larger than those in the driven member, so that the circulatory motion of the oil, and therefore the transmission of power, from the one to the other is always maintained.

Under conditions of maximum acceleration of the engine, the slip dies rapidly away from 600 to 1000 r.p.m. At 1500 r.p.m. it is 8 per cent and at 3400 r.p.m. it has dropped to 2 per cent. Under normal conditions of maintaining uniform speed, the slip is insignificant.

I have read that because I should like to have it commented upon and discussed by our American press; I think it is worth while. At least it should be experimented with in a number of American shops. A big company in England has adopted it, making a large luxurious car, and it seems to me that we should pay a little attention to a device of that sort.

The rubber men should note that rubber has been mentioned two or three times. The remark has been made facetiously that rubber teeth would be advantageous in some of the gears, but that is not so very funny when we consider that wooden teeth have been in use for more than 100 years in some of the gears in the Philadelphia Water Works.

Mr. Strickland, what do you tell the men in the sales department of your company when they say they want a four-speed transmission?

More Power Better than More Gears

WILLIAM R. STRICKLAND[9]:—I would tell them that we do not need it. We are in a little different position than the rest. We have virtually solved that problem by adding more cylinders.

CHAIRMAN LITLE:—Would another speed be equal to eight cylinders?

MR. STRICKLAND:—No, sir, because when the man on the road is driving in high gear he wants a reserve of power and does not want to shift the gears down to a lower ratio to get it. He really wants a combined engine and gear ratio, we may say, that will do the trick. That requires a certain amount of reduction; the heavier the car is, the more engine or the more reduction is necessary. An additional pair of gears will do no good. That is a fact that I suspect has been overlooked. It is a big question, but simplicity favors three speeds if the car has enough power.

I remember having read about the hydraulic transmission in the English *Autocar*. I think it is just a hydraulic clutch. No mention was made of reverse gear or low-speed gear in the extracts that Mr. Litle read, but I think it is somewhere in the article. So it is not a transmission problem, although some are working on a hydraulic transmission that will some day come out if the weight can be brought down to the proper level.

Advocates Two-Driving-Range Transmission

H. W. ALDEN[10]:—I sometimes think that the motor-car engineers are entirely off on the wrong foot in all this argument about three speeds versus four speeds. It seems to me that the public wants two different driving ranges, and I do not believe that the rear axle is the place to have them. If I were given a free hand in designing a transmission system for a motor-car I should select three speeds, but back of the gears I would put a step-down or a step-up transmission, preferably, I think, a step-down transmission with about a 4:1 reduction. That would enable one to have in the rear axle a very fast ratio, and I am a great believer in fast ratios in rear axles for ordinary driving.

CHAIRMAN LITLE:—Do you mean that condition would be better for the axle builder, as it would make a lighter axle?

MR. ALDEN:—I am not thinking of the axle builder, but of the man who builds the engine. He is the one who suffers. I believe the public does not like to shift gears a great deal. If we could have a two-range transmission system in which one range would have, say, a ratio of about 3:1 on the average, and this two-speed device with a ratio of about 1½:1, which would be very simple and take up perhaps 2 in. and certainly not over 3 in. of length in the transmission, we would have one range for traffic conditions and driving in the hills and another range for ordinary touring. Then the middle speed would be used far more than the public uses it today.

I have never driven a four-speed transmission much, although I have to some extent. The public likes to get into one gear-ratio and stay there. I believe that when a driver rolls out of traffic and wants fast acceleration, he should be able to slip out of the two-speed device at the back of the transmission and go along on an entirely different gear-ratio. But I think that the rear axle is not the place for the two-speed device; I believe it is the transmission function. Our company

[9] M.S.A.E.—Assistant chief engineer, Cadillac Motor Car Co., Detroit.

[10] M.S.A.E.—Chairman of the board, Timken-Detroit Axle Co., Detroit.

has done some work with two-speed axles; we have several of them running fairly well, but we cannot help feeling that this is the transmission engineer's problem.

Objection to British Slipping-Clutch

P. M. HELDT[11]:—The British device that Mr. Little mentioned is not a hydraulic transmission, a number of which have been developed in the last few years, but is merely a slipping clutch. It slips under all conditions, for which reason a certain amount of power is always lost. On the large 12-cylinder Daimler that can probably be permitted, because the engine has something like 180 hp.; but when reading the description of the clutch it seemed to me that constant loss of power and the heating would not be wanted on the average car.

Mr. Chase mentioned that if the helix angle of helical gears were comparatively small the gears could be slipped endwise and meshed that way. I should like to point out that if the helix angle is so small that no overlapping of tooth contact occurs the helical gear is not particularly quiet, although a slight amount of quieting action is obtained. The helix angle should be sufficiently large so that each pair of teeth comes into contact before the preceding pair has broken contact.

Plenty of "Punch" with Four Speeds

LOUIS SCHWITZER[12]:—I have been driving cars with four-speed transmissions for three years. The low engine-speed makes driving comfortable, and even at 50 or 60 m.p.h. the driver is unconscious of the engine. Inevitably he will drive at higher car-speeds with more comfort and safety. If he goes back to a three-speed car, he always wants to shift gears when in high. All the four-speed cars give plenty of "punch" at the fourth speed. Ordinary hills can be negotiated very well. On steep hills the four-speed transmission has an advantage over the three-speed, because, if the driver goes back into third speed, it is just enough lower than high speed and is still higher than the second speed of the three-speed transmission so that the car can negotiate the hill at a higher car-speed.

P. M. HELDT

As for simplicity, I think that the four-speed transmission is more simple than the addition of eight cylinders. We should have four-speed transmissions, because I believe that people want low engine-speeds to eliminate noise and because of the fuel economy. I find that those who are driving four-speed-transmission cars are driving faster without feeling that they are working the engine to death. If smaller engines are to be used, the most logical way to obtain better performance from the car would be the application of a supercharger.

MR. STRICKLAND:—One point I should like to bring out regarding the rear-axle gear-ratio with four-speed transmission lowering the engine speed is that it also increases the stresses through the driving train. I have serious doubts about the helical or herringbone gear standing up as well as the spur gear under similar hard conditions. The Citroen company tried many years ago to introduce the herringbone gear cut on one blank, and we tried them extensively. The question of wear arose at that time; it is not easy to get the final accuracy with the treatment that was necessary to enable them to stand up under the stress.

F. C. PEARSON[13]:—Most of us who have designed any transmissions know that the low speed usually has a given ratio in combination with the rear axle, so that the over-all ratio is sufficient to give traction on a muddy road or when the car is off the road in the ditch. We have not been increasing the chassis weight as fast as the piston displacement has gone up, with the increasing size of some of the engines in the present cars, and it seems to me that most of the cars, even with the three-speed transmission, still have as much punch in the low gear as they previously had with the smaller engine. If this is so, I do not see why we need the four-speed transmission. If the same engine were kept and the ratio made faster, there would be a reason for four speeds, but in most cases the engine has been made larger. When that has been done, I believe the sales departments rather than the engineers are responsible for the four-speed transmission.

Driving Torque and Transmission Stress

C. W. SPICER[14]:—Mr. Strickland has called attention to the fact that if we use a faster axle-ratio we increase all the stresses in the transmission line. I reiterate that particularly because the fourth of the eight advantages listed in Mr. Chase's paper may be misleading. He mentioned that the propeller-shaft may be of lighter or less expensive parts. That is true only in some instances in which the reduction in shaft speed might reduce the diameter of the propeller-shaft; that is, over-size that is necessary to prevent whipping. However, it is obvious that the torsional stress in the propeller-shaft as well as in the transmission with, using round figures, a 3:1 reduction-ratio in the axle, will be much higher than it will be with a 4:1 ratio. Consequently, the transmission and all the propeller-shaft parts must have proportionately greater torque-carrying capacity.

MR. STRICKLAND:—One of the strong points that I brought out and illustrated in the paper I read in Detroit on the subject of the extra cylinders in the Cadillac was the question of stress in the driving train. With the reduction in the inertia and gas torque of the 16-clinder engine as compared with the 8, even with the increase in power the peak stress was not increased. The variation in stress was reduced to about one-third what it was in the 8-cylinder car, so the length of life of the driving train will be longer than it was in the 8-cylinder car.

T. L. ROBINSON[15]:—The statement has been made that, "The public wants less gearshifting." While this is undoubtedly true, the public is actually getting more and more of it, due to the wider use of traffic signals, not only in the cities, but in the villages and not infrequently on the otherwise open road. Probably 80 to

[11] M.S.A.E.—Engineering editor, *Automotive Industries*, Philadelphia.

[12] M.S.A.E.—President, chief engineer, Schwitzer-Cummins Co., Indianapolis.

[13] M.S.A.E.—Assistant chief engineer, pleasure-car division, Reo Motor Car Co., Lansing, Mich.

[14] M.S.A.E.—Vice-president, Spicer Mfg. Co., Toledo, Ohio.

[15] M.S.A.E.—Chief engineer, McGill Metal Co., Valparaiso, Ind.

90 per cent of the gearshifting done by the average driver today is in starting from full stops at stop-lights or traffic officers. Since only the three higher speeds of the four-speed transmission are normally in standing starts, and since 90 to 95 per cent of the gearshifting is done from such starts, there would be little difference in the total number of shifts whether a three or a four-speed transmission is used.

HERBERT CHASE

The statement that the public judges a car by its high-speed performance is only partly true, since again acceleration from stop-lights is a much more important factor, mainly because of the greater frequency for the occasion but also because the driver has a direct standard of comparison in the car or cars beside him. Incidentally, a driver will seldom admit that his car was not as well handled as the one next to him which got away faster, but rather blames his car as lacking acceleration. The higher speeds and relatively quiet operation obtainable in the third speed of a four-speed transmission make for rapid acceleration from the standing starts.

E. S. HALL[16]:—As regards getting away from the traffic light, I should like to say that the Spontan transmission beats anything I have ever had the opportunity of seeing. I am not connected with it in any way, but have ridden in New York City traffic in a car equipped with it.

What the Public Wants

HERBERT CHASE:—I gathered from the remarks of the Chairman early in the discussion that he felt I was a little severe, perhaps, in commenting in the paper on the Chrysler transmission. I said it had more than 100 parts, and 13 special tools were required to service it, and that the instructions for servicing covered about 40 pages in a booklet. I merely happened to have the data in that case and did not have it in the others. In justice to the Chrysler company, it should be added that its transmission is not greatly different in that respect from the other internal-gear jobs.

The discussion seems to me to boil down largely to this: that if the car has power enough it does not need more than three speeds. As I see it, the four-speed transmission never was really demanded by the public; it was put in by engineers who thought perhaps it was a good thing, and, to some extent, it probably has been sold to the public through advertising. The public does want performance, and whether it gets it through an engine of adequate size or through a gearset does not matter very much to the car owner, unless an unnecessary amount of gearshifting is required. It seems to me that too much shifting is required with the four-speed gearset. In any case some of those who have tried to get the advantages of a four-speed transmission in respect to reducing the average engine-speed have gone right back to the rear-axle ratios that they had to begin with, or nearly to them, and thereby have sacrificed the gains they set out to capitalize.

I am inclined to agree with what Mr. Alden said, that if we are to have a two-speed device it would be much better to put it on the transmission than on the axle. We have plenty of unsprung weight as it is today, and I think the tendency is going to be away from increases in unsprung weight and probably toward the use of independently sprung wheels.

Mr. Heldt's remarks concerning the helix angle of helical gears are pertinent, but much depends upon what is meant by a small angle. The angle has to be great enough to give an overlap, and depends somewhat upon the width of the gear. It may not be feasible to shift the helical gear as I indicated; I merely wanted to mention the fact that it has been done in transmissions for machine-tools and, if it is desirable, it is not impossible to do it in an automobile gearset. But the helix angle might be so small that the desired quietness could not be obtained.

[16] M.S.A.E.—Automotive engineer, Michell-Crankless Engine Corp., New York City.

Fundamentals of Brake Design

By A. W. Frehse[1]

Semi-Annual Meeting Paper

Illustrated with Diagrams, Drawings and Charts

MODERN brake-development is now reaching a stage in which the performance of a brake design can be predetermined accurately with almost the same certainty as that for powerplants. Accurate mathematical expressions and formulas now supplant the old cut-and-try method used on many types of brake. Armed with these mathematical tools, a designer can proceed with his designs without the usual misgivings and uncertainties that characterized former brake design. In practically every case, the original layout using the principles outlined has withstood the rigors of breakdown testing and field service, with very few modifications of a minor nature.

Thermal and dynamic effects are disclosed, and their influence on the parts constituting the brake are discussed. These effects may spell success or failure in what looks to be a good design. The fallacies of the simple shoe-brake are discussed, and a method for overcoming them is offered.

INTERNAL brakes on modern American cars can be classified generally into three types: (*a*) two-shoe, with common or individual pivots; (*b*) tandem, or interconnected, where a primary shoe actuates a secondary shoe and (*c*) internal, or modified internal band. This paper deals with internal brakes of the two-shoe independent-pivot type only. To present the fundamentals of the other types would require the presentation of subject matter far beyond the limits usually covered in a single paper. It is a matter of a short time when all motor-vehicle brakes will be of the internal type. The unprotected external wheel brakes are rapidly passing out of existence for many well-known reasons.

All internal brakes are self-actuating to a greater or lesser degree, since the point of shoe anchorage is always placed within the confines of the drum chamber. To have a shoe brake with zero self-actuation, it would be necessary to place the point of anchorage at an infinite distance from the center of the drum; or the cam, journal, shoes and drum would have to have absolute rigidity. What is meant by self-actuation is where the rotation of the drum causes the lining-carrying member to increase its normal pressure against the inner surface of the drum, with a given initial pressure, thereby increasing the tangential effort or braking ability; roughly speaking, this is a circular wedging action. This definition might sound unnecessary to those familiar with the principles of brake design. It is mentioned here not only for the benefit of those who are not familiar with its function, but because the principle of self-actuation, its predetermination and control, is a fundamental of efficiency in practically every modern internal brake worthy of consideration.

It is foolish to consider any design of brake without 50 to 70 per cent of self-actuation within the brakes themselves. The average automobile cannot be braked comfortably without considerable outside assistance due to the limited pedal-travel and moderate pedal-pressure required, together with limited hook-up linkage ratios. Many foreign designers are still attempting to use non-self-actuating brakes, and the poor braking ability of their cars bears this out. On some of their high-priced cars, they use separate mechanical servo-mechanisms to increase the shoe pressure. These costly complicated mechanisms are difficult to manufacture and more difficult to control. But why should such mechanical servos be used when there is available a principle of self-servo action within the drum mechanism itself, provided its performance can be predetermined and controlled? It is evident that the more simple the hook-up linkage is, the better and more efficient the braking ability will be. Complicated mechanisms should be eliminated to obtain consistent operation. Therefore, the conclusion has been reached, after years of research and study and by building and testing many different designs, that the proper place for the outside mechanical assistance is within the drum mechanism itself.

Mathematical Principles Involved

In the light of our present knowledge and experience with all types of brake, the two-shoe independent-pivot type is the only one that will lend itself to accurate mathematical analysis and determination. To illustrate more clearly the mathematical principles involved, one shoe of a simple two-shoe brake is shown in the upper left view of Fig. 1. The shoe has engaged with the drum under the initial pressure P, and the rotation of the drum is in the same direction as the shoe has rotated about its anchor pivot. Due to these respective rotations this shoe is the self-actuating shoe. Mathematical proof will not be given at this time that this *is* the self-actuating shoe, as it would detract from the simplicity of what is to be explained.

Let

A = actuation factor for any point on the lining
β_1 = toe angle determined by a diameter through the anchor pin and a radius through the toe of the shoe
β_2 = heel angle
d = distance of the shoe's anchor pin from the center of the drum
f = coefficient of friction of the lining
Q = actuation constant
R = radius of the drum

The actuation factor A can be expressed by the formula:

$$A = \pm f \frac{(R + d\cos\beta)}{d\sin\beta} = Q f \qquad (1)$$

[1] M.S.A.E.—Engineer, Chevrolet Motor Co., Detroit.

Since A = unity for the shoe to be self-locking, $1 = Qf$ or f for self-locking, $= 1/Q$. The factor A is positive when the drum is rotating in the direction shown in Fig. 1, and negative when rotating in the opposite direction. As a concrete example assume the angle $\beta_1 = 30$ deg.; $\beta_2 = 148$ deg.; $R = 5.75$ in.; and $d = 4.25$ in. This gives a lining length of approximately 120 deg., which has been found to be sufficient for all designs and proof of this will be given later.

For the toe,

$$A = f\,\frac{5.75 + (4.25 \times 0.866)}{4.25 \times 0.50} \qquad (2)$$
$$= 4.43\,f$$
$$= Q\,f$$

For the heel,

$$A = f\,\frac{5.75 + (4.25 \times -0.848)}{4.25 \times 0.53} \qquad (3)$$
$$= 0.95\,f$$
$$= Q\,f$$

To be self-locking A = unity; hence,

$$A = 1 = Q\,f = 4.43\,f \qquad (4)$$

Therefore, f, the locking coefficient, $= 1/4.43 = 0.226$. Likewise, for the heel, the self-locking coefficient $= 1/0.95 = 1.05$. This means that if the toe only were bearing on the drum the brake would lock up if the coefficient of friction of the lining is equal to or exceeds 0.226, and is 1.05 at the heel. It is obvious that the heel would never lock up because it is impossible to have a coefficient of friction greater than 1.00. To combine these values for the entire shoe it will be necessary to integrate between these limits; therefore A for the entire shoe is:

$$A = \left(\frac{R}{d}\right) f\,\frac{\int_{\beta_1}^{\beta_2} \sin\beta\,d\beta + (d/R)\int_{\beta_2}^{\beta_1} \sin\beta\,d\beta\cos\beta\,d\beta}{\int_{\beta_2}^{\beta_1} \sin^2\beta\,d\beta} \qquad (5)$$

$$\int_{\beta_1}^{\beta_2} \sin\beta\,d\beta = -(\cos\beta_2 - \cos\beta_1) \qquad (6)$$
$$= -(-0.857 - 0.866)$$
$$= 1.723$$

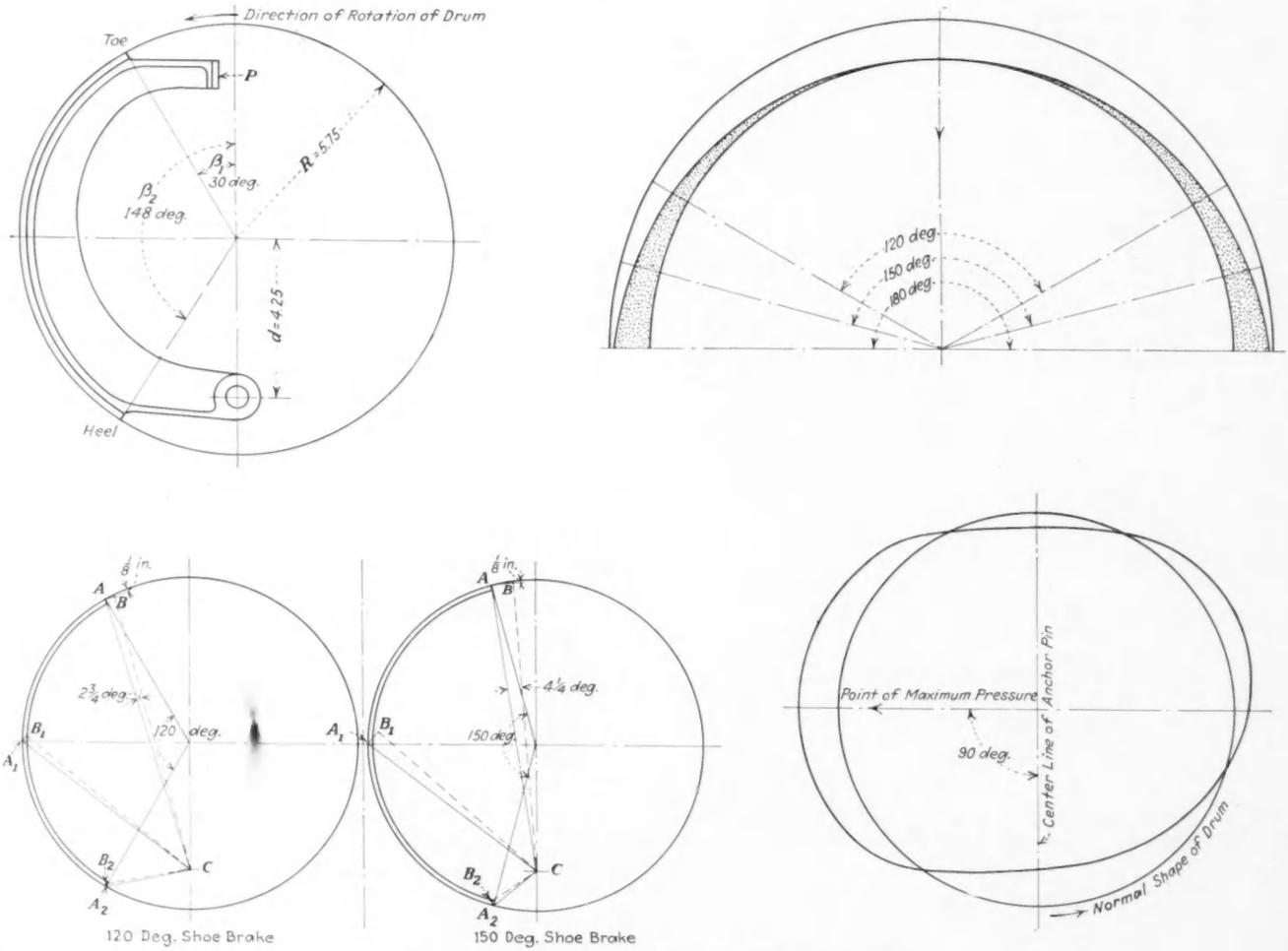

FIG. 1—ANALYSIS OF A SIMPLE TWO-SHOE BRAKE

In the Upper-Left View the Shoe Has Engaged with the Drum under Initial Pressure P and the Rotation of the Drum Is in the Same Direction as the Shoe Has Rotated about Its Anchor Pivot. Three Lengths of Shoe Respectively 180, 150 and 120 Deg., Are Shown in the Upper-Right View. The Inner Circle Represents the Face of the Shoe; the Outer, the Drum; and the Space between, the Lining Thickness, Which Has Been Purposely Exaggerated To Illustrate the Principle More Clearly. The Linings of Two Simple Shoes, One of 120 and the other of 150 Deg. Are Compared in the Lower-Left View, Both Having the Same Mathematical Characteristics. At the Lower Right, Drum Distortion Is Illustrated. It Is Shown Greatly Exaggerated. It Sometimes Amounts to as Much as 0.02 In., According to the Intensity of the Braking

$$\int_{\beta_1}^{\beta_2} \sin\beta\, d\beta \cos\beta\, d\beta = -1/4\,(\cos 2\beta_2 - \cos 2\beta_1) \quad (7)$$

$$= -1/4\,(0.438 - 0.500)$$
$$= 0.0154$$

$$\int_{\beta_1}^{\beta_2} \sin^2\beta\, d\beta = (1/2\,\beta_2 - 1/2\,\beta_1) - 1/4\,(\sin 2\beta_2 - \sin 2\beta_1) \quad (8)$$

$$= -1/4\,(-0.899 - 0.866)$$
$$= 1.4812$$

Substituting values in equation (5) we have,

$$(5.75/4.25)\, f \frac{1.723 + [(4.25/5.75) \times 0.0154]}{1.4812} = 1.58\, f \quad (9)$$

The theoretical locking coefficient for the entire shoe is, therefore, $1/1.58 = 0.63$. This means that the shoe will become self-locking if the coefficient of friction is equal to or exceeds 0.63, regardless of the initial pressure P. Therefore, a lining should be selected whose coefficient must never equal 0.63.

Equations (1) to (9) deal with a shoe known as being of an unsymmetrical type; that is, where a line drawn from the center at 90 deg. to a diameter through the anchor pin does not divide the shoe into equal segments. A symmetrical shoe is, by the foregoing method,

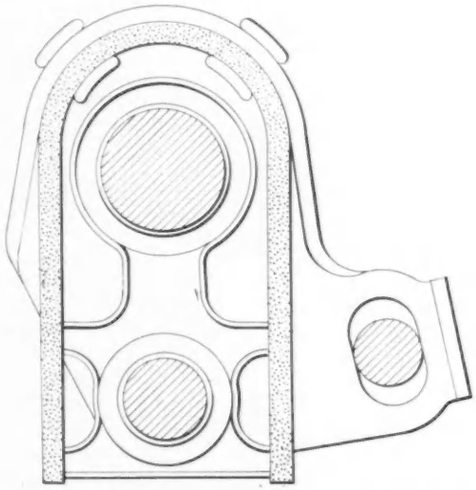

FIG. 2—DEVICE TO PERMIT CLOSE CLEARANCES BETWEEN LINING AND DRUM

A Floating Cam, Together with What Is Called a Spring-Mounted Centralizer Was Developed as Shown in the Cross-Section through the Camshaft, Which Is Free To Drift Left or Right, about Its Pivot under Constraint of the U-Shaped Spring

divided into two equal segments. When computing for a symmetrical shoe the second expression in equation (5), $\int_{\beta_1}^{\beta_2} \sin\beta\, d\beta \cos\beta\, d\beta$, becomes 0 and therefore drops out.

Assume that a lining is available that has all the desirable characteristics demanded by a brake and whose coefficient of friction averages 0.43. It has been found from experience that the computed locking-coefficient should be at least two points in the first decimal place higher than the high limit of the coefficient of the lining. With the diameter of the drum and the length of

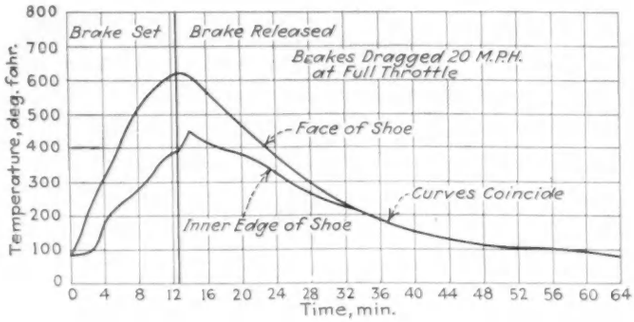

FIG. 3—TEMPERATURE RISE IN THE BRAKE SHOE

When Cooling, Various Parts of the Shoe Gradually Assume the Same Temperature but This Only Occurs After the Brakes Have Been Released

the shoe determined, the distance of the anchor pin from the center is readily computed by the formula:

$$d = \frac{r \int_{\beta_1}^{\beta_2} \sin\beta\, d\beta}{Q \int_{\beta_1}^{\beta_2} \sin^2\beta\, d\beta - \int_{\beta_1}^{\beta_2} \sin\beta \cos\beta\, d\beta} \quad (10)$$

$$= \frac{5.75 \times 1.723}{(1.587 \times 1.4812) - 0.0154}$$
$$= 4.25$$

Thus, from equation (10), it is easy to lay out the important points of any simple shoe-brake of predetermined characteristics, whether the lining coefficient is given or if it is desired to determine what coefficient of friction to specify with a given design. The amount of self-actuation of the brake would then be expressed by the ratio of $f/(1/Q)$ which, in the foregoing case, would be 68 per cent.

Desirable Length of Shoe Analyzed

It was stated that 120 deg. of shoe length was sufficient for any design of shoe brake. Some engineers are of the opinion that the longer the lining of a shoe is, the longer the life will be. This would be true if the normal pressure against the lining were equal at all points. But it does not apply in the case where the

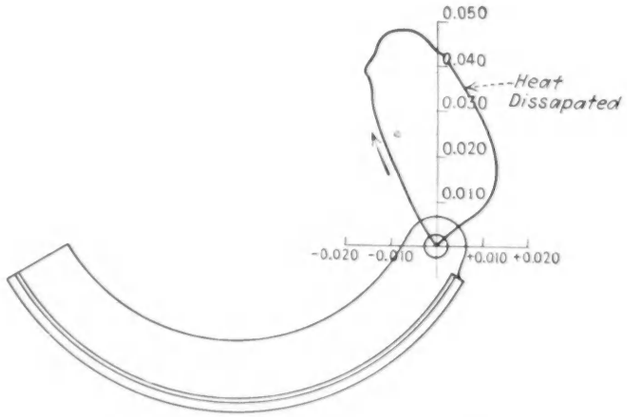

FIG. 4—EFFECT OF HEAT ON THE SHOE

The Shoe Curls Up as Shown on Account of the Greater Expansion of the Metal, the Orbit of the Eye-End Moving Clockwise. This Curl Is Resisted Greatly by the Cam Pressure but Not Sufficiently To Keep the Shoe in Its Normal Shape

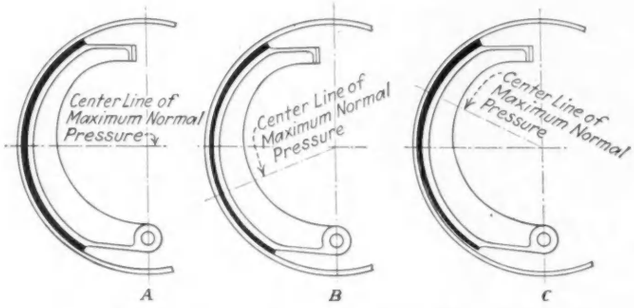

FIG. 5—DIAGRAMS ILLUSTRATING WEARING ACTION

Diagram *A* Shows the Shoe in Its Normal Shape in Contact with the Drum. Diagram *B* Shows the Wearing Action Taking Place at the Point of Maximum Pressure on the Curled Shoe Due to This Uneven Heat. Diagram *C* Shows How, After the Shoe Itself Becomes Cool and Returns to Its Normal Shape, the Lining Is Thinner at the Heel Than at the Toe and How the Next Application Will Cause the Shoe To Make Contact at the Toe Only

initial pressure is applied at the free end of the shoe. Many attempts have been made to secure equal normal pressures all around; but, so far, they have proved to be unsuccessful.

Three lengths of shoe, 180, 150, and 120 deg., respectively, are shown in the upper right view in Fig. 1. The inner circle represents the face of the shoe; the outer, the drum and the space between, the lining thickness, which has been purposely exaggerated to illustrate the principle more clearly. If the outer circle is moved toward the inner, representing lining wear at the point of maximum normal pressure, the shaded portion then represents unused or excessive lining. Due to the flexibility of the drum and shoe, this does not hold quite true; but the principle applies and can be proved by measurements from actual service. We have many cars that have traveled over 40,000 miles of brake test schedule on the original 120-deg. liners, and that have carried 450 lb. of ballast, exclusive of driver and equipment.

A longer shoe requires more cam travel and consequently more pedal stroke with a given mechanical ratio. The linings of two simple shoes, one of 120 and the other of 150 deg., are shown at the lower left in Fig. 1; but both have the same mathematical characteristics. For a given clearance at the point of minimum clearance, which is at the toe, and in this case was made a large amount so as to illustrate the principle more clearly, it is apparent, by considering geometric motion from A to B, A_1 to B_1, and the like, that the 150-deg. shoe must travel $4\frac{1}{4}$ deg. as compared with the $2\frac{3}{4}$ deg. for the 120-deg. shoe, which is a ratio of about $1\frac{1}{2}$ to 1. This ratio applies to any given clearance. Therefore, if it requires a 2-in. pedal-stroke to set the brake with the 120-deg. shoe, it will require a 3-in. stroke to set the 150-deg. shoe, the mechanical-advantage ratios being the same in each case. It is necessary to conserve the pedal stroke as much as possible, because it already is limited enough for comfort, and it is desirable to take-up for wear as infrequently as possible. Other additional detrimental factors—such as loss of pedal travel due to heat expansion of the drum, mechanical distortion caused by shoe pressure, fade-out on account of shoe curl which necessitates higher pedal-pressure and hook-up deflection commonly known as "rubber"—subtract from the available pedal-travel. A 120-deg. shoe is lighter, cheaper and easier to manufacture, and a more uniform contact of the lining against the drum is possible.

Drum Distortion Considered

When pressure is applied to the free ends of the shoes, the forward-acting shoe illustrated at the upper left in Fig. 1, being the one having the positive actuation, will exert a greater normal pressure, $P + A$, on its side of the drum and cause it to distort considerably. This distortion, shown greatly exaggerated in the lower right view in Fig. 1, sometimes amounts to as much as 0.02 in., according to the intensity of the braking. The measurements were taken when the drum was rotating. It should be noted that the point of maximum drum-deflection is at the point of maximum shoe-pressure, which is always on a line 90 deg. from a diameter through the anchor pin, provided that the shoe has not distorted.

To maintain the initial pressure on the free end of the shoe, the expanding means must be able to follow up this forward drift of the shoe into the drum, taking its reaction against the reverse or negative-acting shoe. If the expanding means, such as a cam, is rigidly journaled in the brake support to prevent this follow-up, then the cam describing equal motion only will exert its pressure on the negative-acting shoe, its reaction against the opposite walls of the journal, and the resultant pressure on the forward-acting shoe cannot build up. This action is not so serious when the two shoes have exactly the same clearance, the drum and shoe do not distort appreciably, and the forward-acting shoe will not wear down faster than the reverse. But this condition will not exist very long and the shoes must be constantly readjusted.

Foreign brakes and their efficiencies are typical examples of the foregoing fallacy. The constant readjusting on this type of brake is so evident that some foreign cars have brake adjusters located so that they can be reached without leaving the driver's seat. This detrimental action led to the development of the floating cam, mounted in such a way as to permit the cam

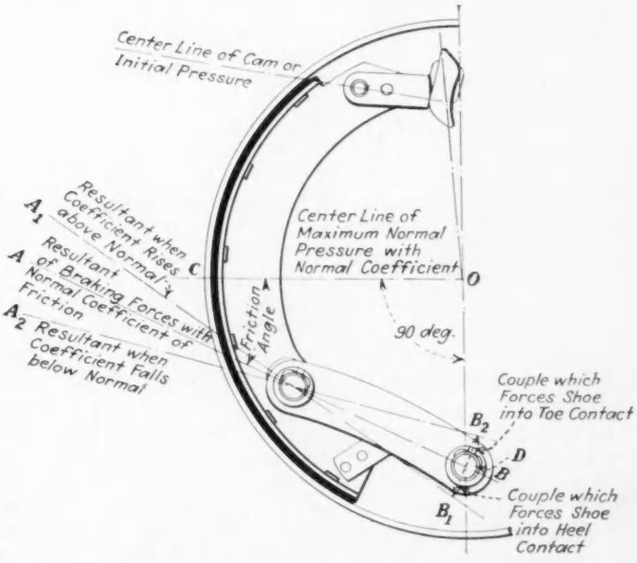

FIG. 6—THE SELF-ACTUATING SHOE AND ITS LINK

The Resultant of the Braking Forces Must Pass through This Link and, under Normal Braking-Conditions, This Resultant Is Precisely on the Center Line of the Link Shown

FIG. 7—SHORT REVERSE-SHOE

The Short Shoe Is Shown in Position with Respect to the Forward Shoe. The Cam Curves Are Determined Graphically on Layouts Several Times That of Full Size so as To Secure Great Accuracy and Are Designed To Give a Constant Leverage-Ratio

to follow up the forward-acting shoe. However, the mere floating of the cam does not entirely fulfill the requirements. After the brakes are released, the cam must be returned positively to its original geometric center between the shoes, or the forward distorted portion of the drum after a heavy brake-application will act as a return spring against the shoe and cause it, and the cam, to move back the exact amount of this distortion. Then, after the drum shrinks due to cooling off, the brake drags unless independent retractor springs are used.

To obviate brake drag a floating cam, together with what is called a spring-mounted centralizer, was developed as illustrated in Fig. 2. It shows a section through the camshaft, which is free to drift, left or right, about its pivot under constraint of the U-shaped spring. The spring exerts an initial pressure of about 50 lb. on either side of the cam journal so as to overcome any tendency to stick on account of lack of lubrication or rust. The spring is held in a separate carrier which is adjustable, to bring the cam originally to the exact geometrical center between the ends of the shoes, and then is clamped to the backing plate.

The adjustment to this center is very simple; it is necessary only to exert pressure on the brake pedal; then the cam, the journal and its carrier automatically will centralize themselves on account of torque balance. Then, when the brake is applied, the cam is free to move in the direction of the distortion under the constraint of the spring, but the drifting pressure is greater than that exerted by the return spring due to the leverages. When the brake is released, the cam is returned to its original position, and the brake will not drag after cooling off.

When discussing equation (1), the actuation of the simple shoe, the locking coefficient was determined for the toe and heel separately. This was done to bring out the importance of full arc-contact of the shoe against the drum, the lack of which will cause a wide variation in the brake performance all the way from grab to total fade-out. This very range of performance will occur in a simple shoe-brake due to shoe curl or warp. When the brakes are applied, the resultant frictional forces cause a considerable rise in temperature of the braking parts, for the car energy must be transformed into heat.

Heating Effects Discussed

The temperature rise in the shoe is shown in Fig. 3. It will be noted that, when cooling, various parts of the shoe gradually assume the same temperature, but this only occurs after the brakes have been released. The outer periphery of the shoe is hotter than the inner by some 150 deg. fahr., actually measured on the road by means of thermocouples. Due to the greater expansion of the metal at this place, the shoe curls up as shown in Fig. 4, the orbit of the eye-end moving clockwise. This curl is resisted greatly by the cam pressure but not sufficiently to keep the shoe in its normal shape.

Diagram *A* in Fig. 5 shows the shoe in its normal shape in contact with the drum. Diagram *B* shows wearing action taking place at the point of maximum pressure on the curled shoe due to this uneven heat, which has been exaggerated for the sake of clearness. After this curling has taken place most of the braking is being done toward the heel of the shoe, where the actuation is less and causes fade-out. Because of the high pressures necessary to stop the car, the lining at the heel is rapidly worn away. After the brake is released the shoe cools off and returns to its original shape, but not so with the lining. It is now thinner at the heel than at the toe and the next application will cause the shoe to make contact at the toe only as shown by diagram *C*.

On account of the high actuation at the toe as previously computed, the brakes will lock, or grab, until this

FIG. 8—COMPLETED REAR-WHEEL BRAKE WITH SEPARATE AND INDEPENDENT EMERGENCY BRAKES DESIGNED ENTIRELY FROM THE PRINCIPLES STATED IN THE PAPER

portion is worn down to the level of the heel, or until the shoe has curled up again. This alternation of heel and toe contact known as fade-out and grab will repeat itself indefinitely, unless steps are taken to eliminate it. Drum and shoe flexibility will mitigate it to some extent, but not enough to overcome the objection entirely. Very low coefficients of linings also help, especially when combined with flexible drums and shoes, but these cause high pedal-pressures and "rubbery" feeling brakes. In some designs this condition reaches uncontrollable proportions, especially where very rigid drums and shoes are used. To overcome these undesirable characteristics of a simple shoe-brake, a link is interposed between the shoe and its anchor pin. The angular position of this link is critical and is determined mathematically. Due to the articulation of the shoe, the computed theoretical locking-coefficient is modified to what is called the actual locking-coefficient.

Study of Self-Actuating Shoe and Link

The value of the actual locking-coefficient is determined by the lining under consideration, its range of coefficient and the directions of the lines of the initial and resultant forces. In practice its value is assumed to be about three to four points in the first decimal place above that of the mean coefficient of the lining. The difference of this actual locking-coefficient over that of the lining then represents a safety factor to take care of variation.

Fig. 6 shows the self-actuating shoe and its link. The resultant of the braking forces must pass through this link and, under normal braking-conditions, this re-

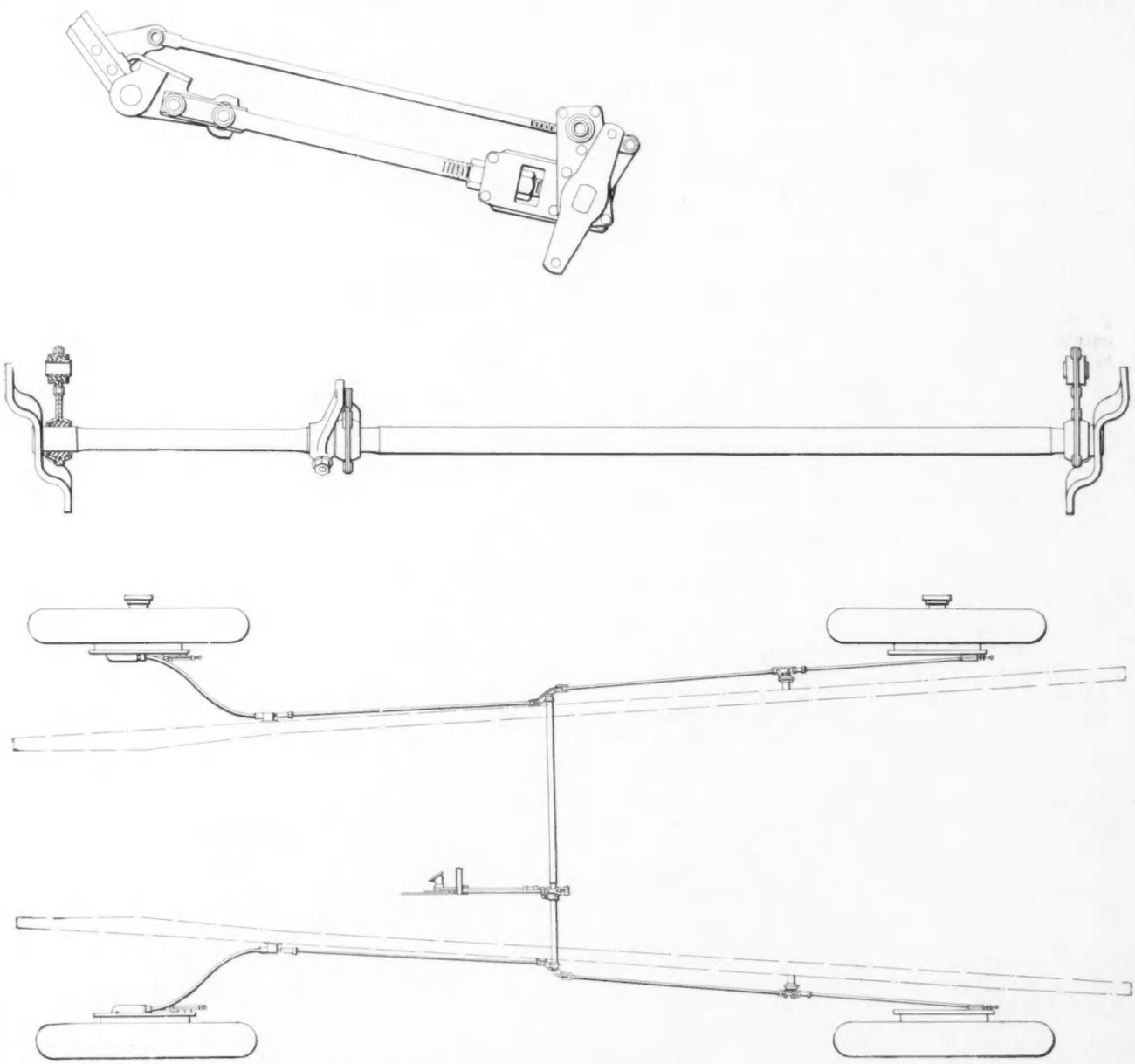

FIG. 9—CROSS-SHAFT ILLUSTRATING IMPORTANT FACTORS IN MECHANICAL BRAKE HOOK-UP SYSTEMS (UPPER); AND PLAN OF THE COMPLETE SERVICE HOOK-UP MECHANISM (LOWER), SIMPLICITY AND DIRECT PULL BEING THE FEATURES KEPT IN MIND

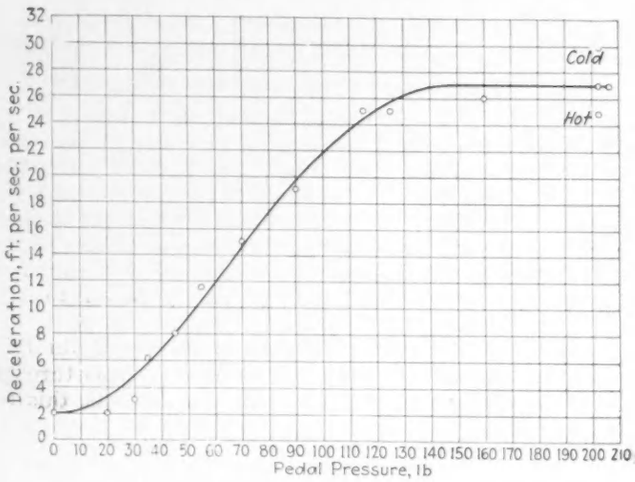

FIG. 10—DECELERATION CURVE CONSIDERED BY MANY DRIVERS TO BE IDEAL

sultant is precisely on the center line of the link shown. Then the shoe operates the same as a simple shoe, moving about its anchor-pin center during setting and releasing of the brake. If the link is not exactly on the resultant of the braking forces, a couple is set up, the shoe will move about the articulation point and improper braking action will result.

With the angular position of the link correctly computed no movement occurs during normal brake-application and, when the shoe curls under heat, the ends being free to move, contact is maintained at the area of maximum pressure which is in the center; the leverage arm remains unchanged and the efficiency does not fall off appreciably. The maximum-wear point is likewise at the center of the shoe; no excessive heel or toe wear occurs and thus alters the contact of the shoe in the drum which, as already pointed out, very materially affects the actuation.

If the coefficient of the lining should rise, the resultant of the braking forces will not be on the center line of the link $A\ B$, but will take a direction as shown by the line $A_1\ B_1$, due to the increase of the friction angle, and the couple $D\ B_1$ tends to force the shoe into heel contact and to decrease the actuation. Conversely, if the coefficient should fall off, since the friction angle is less, the couple $D\ B_2$ forces the shoe into toe contact where the actuation is greater. Thus it can be seen that the action of the link tends to keep the actuation constant, regardless of change of coefficient, and uniform braking is the result.

It must be understood that the foregoing principle will not take care of a range of coefficient such as from 0.3 to 0.8, but will compensate for a range of coefficient from 0.4 to 0.6. Laboratory and road tests completely verify the action of this shoe and link, and the wear diagrams on the linings check almost exactly with the computed theoretical determinations. It is a great source of satisfaction to observe how closely the actual performance follows the theoretical. To assure that the shoe moves about its anchor pin only during application and release, friction is introduced between the shoe and its link; otherwise, it would be impossible to set the shoes to close clearances.

There is ample evidence that in any two-shoe brake, the forward-acting shoe is the self-actuating shoe and consequently does about 75 per cent of the braking. The reverse-acting shoe becomes the self-actuating shoe when the car is reversed. As reverse braking is such a small percentage of that of forward braking, the life of the reverse-acting shoe is more than twice that of the forward shoe. For this reason the reverse-acting shoe is made shorter, usually about 60 deg. of total length. The two linings will then tend to wear down at the same rate. It actually could be made still shorter, but its pressure would then cause too much distortion of the drum at its area of contact. Shortening this shoe saves lining cost and shoe-material cost and reduces the weight of the mechanism appreciably. Incidentally, the remaining space inside the drum can be utilized for a separate two-shoe emergency-brake.

This short reverse-shoe is placed so that the toe is approximately symmetrical with that of the forward-acting shoe about the center line of the cam or expanding mechanism. Its actuation is computed separately from that of the forward-acting shoe, using the same formulas, and is calculated so as to give a light pedal-pressure for reverse braking. The locking coefficient can be safely brought down to be almost the same as the coefficient of the lining. On account of its short contact-length, long radius and the position of its anchor pin, it requires by geometric motion less cam travel for a given drum clearance. By proportioning the effective lever-arm lengths of the cam according to the respective shoe-travels, since the cam is able to float, great saving of pedal travel is secured.

Fig. 7 shows this short shoe in position with respect to the forward shoe. The cam curves are determined graphically on layouts several times that of full size to secure great accuracy and are designed to give a constant leverage-ratio. The procedure for determining the curves is very similar to that of engine-cam design.

The single retractor spring is placed so that its line of pull intersects the point of intersection of the link-angle and cam-force lines produced. Then the pull of the retractor spring will not affect the articulation of the forward-acting shoe. All this attention to detail analysis may seem superfluous; but taking an engine as an example, without such analysis to the smallest details, the modern engine would never be what it is today. Many times, modern brakes are called upon to do five times the work an engine can do in a given length of time.

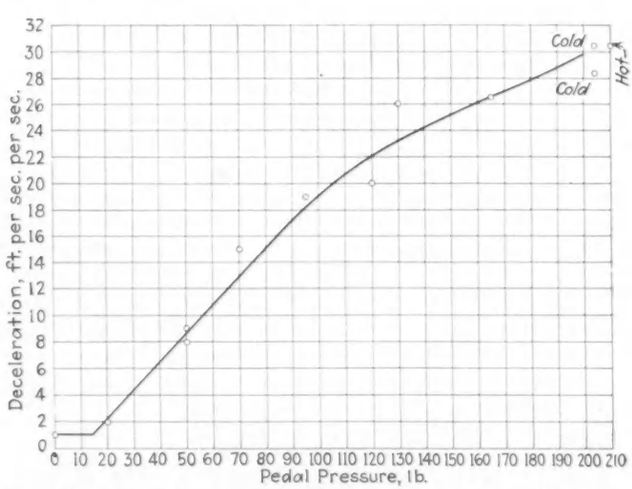

FIG. 11—CURVE OF THE SAME CAR AS FOR FIG. 10 AFTER 42,000 MILES OF SERVICE ON THE ORIGINAL BRAKE-LININGS

Fig 8 shows a completed rear-wheel brake with separate and independent emergency brakes designed entirely from the foregoing principles. With few exceptions, the parts are made entirely of pressed steel. The fronts are exactly similar with the exception of the emergency-brake mechanism. The cam levers and adjusting screws of this particular design are placed outside of the brake proper for the sake of accessibility and quick take-up for wear. As there are no cover plates to remove for adjusting and the cams are of uniform rate, I have taken up or adjusted a set of these service brakes in about 3 min. on the road with one tool without crawling under the car. Factory adjustments on the assembly line require approximately 4 to 8 sec. per car, with one man at each wheel.

Mechanical Brake Hook-Up Systems

Very careful analysis must be given to the chassis hook-up system of a mechanical brake. Geometrical motions must be perfect. Friction must be reduced to the minimum and must remain uniform. Deflections can be reduced to the minimum by using straight pull-rods with no offsets, and bending and torsional moments can be reduced by properly proportioning the parts. The upper view in Fig. 9 illustrates a cross-shaft as an example of bearing these comments in mind. The pull-rod from the pedal is straight and, because the pedal is at the left of the car, the input lever on the cross-shaft is not on the center line. This tends to produce unequal pressures on the outside levers. To obviate this, the shaft is necked down on the short side so that its polar moment of inertia is exactly proportional to the length. Forward bending of the cross-shaft is eliminated by supporting it directly in line with the pull by means of a strut parallel to the pull-rod.

By supporting the outer ends on floating journals, the levers are permitted to equalize front and rear pressures for more uniform braking. By equalizing these pressures left and right and front and rear, the life of the linings was increased over two-fold. An inherent advantage of a first-class mechanical hook-up system is that the brakes will always tend to equalize themselves. Cars in many instances have been observed to pull slightly to the right or left with a new set of liners. After driving 100 miles or so with moderate braking, the brakes equalized themselves and remained so. Extreme pulling to right or left, of course, must be adjusted out. The journals are non-metallic and self-lubricating, and the journal brackets are mounted on rubber bushings to reduce friction from frame distortion and eliminate rattles.

Fig. 9 shows in the lower view the plan of the complete service hook-up mechanism, simplicity and direct pull being the features kept in mind. Cable and conduit are used for the front brakes on account of the perfect geometry obtained, light and clean-cut construction, absence of deflection and safety of control. The hook-up pull-back spring is placed at an angle in the horizontal plane, so that the lateral component of its pull keeps the end-play of the cross-shaft from rattling.

Hook-up linkage ratios should be selected to give about 18 to 22 ft. per sec. per sec, deceleration at a 100-lb. pressure on the pedal. A 70-per cent self-actuation in the brakes and a total mechanical advantage of 60 to 1 from the pedal to the shoes will give this performance provided deflections and friction are properly taken care of. Such a set-up will give decelerations as high as 30 feet per sec. per sec. even after 40,000 miles of service. It requires adjustment for wear about every 10,000 to 12,000 miles in ordinary service, if the size of the brakes has been worked out in proportion to the weight of the car.

Fig 10 illustrates a deceleration curve considered by many, including women drivers, to be about ideal. The slope of the curve is about 45 deg. when plotted as shown, and the curve should be a straight line up to 18 to 20 ft. per sec. per sec., showing that the deceleration is proportional to the pedal pressure, which is believed to be proper. Beyond 20 ft. per sec. per sec., the slope of the curve should fall off to eliminate the danger of over-braking in an emergency. On ordinary good gravel-roads, 18 ft. per sec. per sec. deceleration can be comfortably used for maximum stopping; and on dry concrete and asphalt, 20 ft. per sec. per sec.

Fig. 11 illustrates a curve of the same car after 42,000 miles of service on the original brake linings. This performance illustrates the possibilities of what can be done with properly designed internal brakes, together with a good lining and efficient hook-up system.

Brake-Linings Are Improving

The lining manufacturers have made considerable progress in the last few years on linings that are beginning to have some semblance of a uniformity of coefficient of friction. Any brake, no matter how finely analyzed, computed and designed, will be a failure if used with poor lining. The characteristics of the lining used absolutely control the characteristics of the brake performance. It is impossible, with reasonable constructions, to seal the brakes against all outside disturbing factors. There must be some clearance between the rotating and stationary parts, and through this space moisture and some foreign matter are bound to enter. These increase the difficulties with both the lining and the car manufacturers in developing better linings and brakes. There are many problems such as: obtaining a uniform coefficient of friction, proof against moisture and temperature changes, as well as providing non-scoring properties and ability to withstand high unit-pressures with low wear-factors.

Another problem is the disposal of the worn-off lining material, commonly known as powder. This substance absorbs moisture very readily due to its powdery state. When the car is in motion, the powder not scraped off from brake application is distributed over the inner face of the drum by centrifugal force. When the car is stopped, that which is on the upper surface drops down on the lining faces and absorbs moisture. The next time the brakes are applied the coefficient of the lining has momentarily been changed. This may not be so serious, but it does affect uniformity of brake performance. I firmly believe that the time is not far off when linings can be specified with the certainty and confidence that apply to modern alloy-steels today.

In conclusion, I take no credit for the majority of the principles presented and acknowledge the efforts of the engineers who have devoted many hours of hard work and diligent study in working out the mathematical and operating principles involved.

THE DISCUSSION

A. Y. DODGE[2]:—I wish to call to attention the fact that drum warping is almost opposite to the natural movement of the shoe or what one might call the natural lining-wear diagram; they tend greatly toward offsetting each other. After looking at thousands of shoes whose wear diagrams show that more wear should take place in the middle of the shoe than at the end, we find that the wear is almost uniform if an average of a large number of shoes is taken; that is, when the drums are properly proportioned to the shoes.

Mention has been made of a shoe which changes its shape slightly with heat, and a drum which changes its shape with pressure. These changes tend to counteract each other. We feel that a brake having proper shoe-construction and proportion will blend those two factors together. A simple form can accomplish much of what Mr. Frehse has accomplished by using links.

CHAIRMAN W. R. STRICKLAND[3]:—Does that apply to the same design of shoe or to some other type?

MR. DODGE:—We should confine the discussion to the subject of a simple shoe as shown, self-actuating, with the drums revolving in the same direction as the shoe is revolved, and compare this with the simple shoe shown which is mounted with links.

Tire Wear Considered

B. J. LEMON[4]:—The point brought out that the brake must absorb five times the power at some particular period that the engine can produce in that same period is interesting to the tire industry, I think. Does Mr. Frehse think that, over a run of a number of thousands of miles, general running in traffic and on the road, there is more power required to stop the car through the tires than to accelerate the car, keeping in mind that this ratio of 5 : 1 may be brought into play in a number of cases for deceleration; that is, are tires being worn off more by deceleration than by acceleration, neglecting the regular run between the starting and stopping?

A. W. FREHSE:—It is difficult to make measurements like that. The tires stand up fairly well in spite of the brakes. The acceleration force is much less than that of retardation. The best acceleration one gets with a car in high gear is about 4 ft. per sec. per sec. Of course, brakes in some instances will easily develop as high a deceleration as 30 ft. per sec. per sec.; but, normally, about 20 ft. per sec. per sec. is the usual braking for quick stopping, driving up to the light and stopping a reasonable distance from a speed of 25 m.p.h. The tires seem to withstand the service all right.

CHAIRMAN STRICKLAND:—I remember examining many tires that would illustrate too much power or too much braking such as too much wear on the tread, for instance, too much side slippage and things of that kind. If the braking were more severe, I think that it would have shown up in that way. I do not mean slipping on the pavement, but refer to the pull of the rubber, which one can generally see. Even with the increase in the braking, I think that still the natural wear and tear due to the driving predominates.

W. C. KEYS[5]:—I cannot agree with your interpretation of Mr. Lemon's question. I think he confined his question to the difference in wear on tires in acceleration versus deceleration, neglecting driving conditions. In any of the cities with which one is familiar, one can see more rubber dust on the near side of the traffic light where cars must stop than on the far side where the cars are accelerating. In my opinion there is a very considerably greater wear from applying brakes than from accelerating.

CHAIRMAN STRICKLAND:—That raises the question whether the cars are using these four-speed gearboxes or not, and whether they have power enough. That condition will vary from year to year depending on the ratio of power to brakes.

BURNS DICK[6]:—I am glad to note that Mr. Frehse has gone into the question of drum distortion. This matter was mostly passed up in previous papers read before the Society and, without it, the mathematical discussions alone may lead to incorrect conclusions.

I believe that the question of squeak is very largely tied up with distortion and that it is partly because distortion is not uniform that squeak is so elusive. Some development work which we recently carried out shows that drum distortion coincides very closely with the diagram shown in Mr. Frehse's paper, except that we have gone somewhat farther and have taken several positions crosswise of the drum to determine the relative distortion at the closed end and the open end, as well as around the circumference.

The distortion for the forward shoe is greater in magnitude than for the reverse shoe, as would be expected; but, for both the forward and the reverse shoe, the amount of distortion increases toward the open end of the drum, in spite of the reinforcing flange which is usually provided at the open end. Therefore, if a reinforcing ring is shrunk onto the drum to prevent distortion, it should be located toward the open end rather than in the center.

JOHN WHYTE[7]:—I think the point about drum distortion at the open end is very well brought out in some experiments on large truck-brakes where they are using 4 or 5-in. widths. Examination of the liner on those brakes under severe conditions will show that the maximum wear is on the inner edge all the way around, simply due to the coning of the drum.

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[2] M.S.A.E.—Consulting engineer, Bendix Aviation Corp., South Bend, Ind.

[3] M.S.A.E.—Assistant chief engineer, Cadillac Motor Car Co., Detroit.

[4] M.S.A.E.—Field engineer, United States Rubber Co., Detroit.

[5] M.S.A.E.—Chief engineer, automotive development department, United States Rubber Co., Detroit.

[6] M.S.A.E.—Chief engineer in charge of automotive engineering division, Wagner Electric Corp., St. Louis.

[7] M.S.A.E.—Chief engineer, Warner Electric Brake Corp., Beloit, Wis.

Electrically Operated Vehicle Brakes

By JOHN WHYTE[1]

Semi-Annual Meeting Paper

Illustrated with Photographs and Drawings

USERS of motor-vehicles are demanding more positive brake control with the exertion of minimum physical effort. This is particularly true in the commercial field, where large loads must be handled and comparatively high speeds are being obtained.

Because of definite limitations in the amount of self-energizing that is obtainable in existing brake systems, an increase in brake output is desirable beyond that obtainable by accepted methods of self-energizing and with the use of reasonable pedal pressures.

Boosters are of assistance in this respect. Electrical energy may be used as a means of energizing brakes, making it possible to obtain any predetermined brake output without effort on the part of the operator. The electrical system offers advantages due to the ease of electrical distribution and simplicity of installation.

Discussion brought out some details of the mechanical actuation of the electrical brake and questions of brake cooling and transfer of heat to the tire. Attention also was given to the limitations of self-actuation that can be employed without seizing; to adjustment of the brake for intensity of effort and distribution between front and rear wheels; to brake equalization; and to brake squeaks, which the author says are caused by some mechanical weakness or looseness.

JOHN WHYTE

AUTOMOTIVE-vehicle brakes have been the subject of a great deal of attention, with the object of increasing the performance efficiency of this part of the vehicle equipment. By performance efficiency is meant the ability to obtain the maximum performance from the brakes with the least exertion of pedal pressure on the part of the operator.

A study of the basic forces involved in brake operation makes possible a clearer consideration of the entire problem. A vehicle weighing with passengers 5000 lb. and traveling at 60 m.p.h. requires a retarding force of 3420 lb. to stop within a distance of 175 feet. The average deceleration of the vehicle under this condition is 22 ft. per sec. per sec., which may be regarded as a satisfactory rate of stopping.

Assuming an even distribution of stopping force on the four wheels, a retarding force of 855 lb. is required at the tread of each tire. With a 32-in. tire and 16-in. brake-drum, the equivalent torsional force at the brake-drum surface is 1710 lb. If the brake were not self-energizing, the radial force necessary to produce this retarding force would be 4270 lb., where the brake lining has a coefficient of friction of 0.4. Considering the four brakes of the vehicle, four times this pressure, or 17,100 lb., must be furnished by the operator.

The possible mechanical advantage that can be used on brake mechanisms is necessarily limited; otherwise, pedal travel becomes unreasonable and frequent adjustment becomes necessary to take up the wear in the brake-shoe liners.

With a possible leverage between the immediate brake mechanism and the brake pedal of say 40 to 1, a pedal pressure of 427 lb. would be required.

Consideration of these figures shows why self-energizing, servo operation or wrapping must be utilized in brake design. Self-energizing is a legitimate means of increasing the performance of vehicle brakes. There are, however, limits to the use of self-energizing or wrap, the effect of which is a combined function of the arc of contact of the liner and the coefficient of friction of the liner.

Depending on friction characteristics, self-energizing is readily affected by variations in these friction conditions, and commercial brake-lining is an extremely variable product from the standpoint of friction. As a result, the variables existing in the performance of the friction unit make themselves evident in the performance of the entire brake installation, since the variable friction factor is amplified in direct ratio to the self-energizing factor.

The effect of self-energizing is best understood by a comparative example. Consider a full-wrap internal brake having an arc of contact of 330 deg. Were it possible to apply this brake with entire freedom from self-energizing, an applied pressure of 500 lb. would result in a torsional retarding force at the drum surface of 1250 lb. Under the same conditions, but allowing full self-energizing of the brake-band, the force at the same point would be 4500 lb., 3.6 times as much.

Wrapping Effect Multiplies Variations

Should there be any great amount of variation in the friction characteristics of the liners of two brakes on the same axle, the variation would be amplified in the

[1] M.S.A.E.—Chief engineer, Warner Electric Brake Corp., Beloit, Wis.

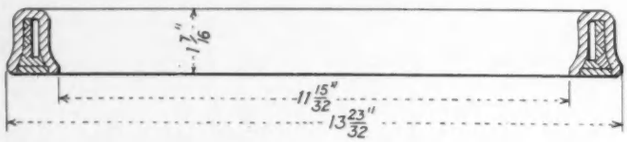

FIG. 1—CROSS-SECTION OF MAGNET USED IN ELECTRIC BRAKE
A Magnet of the Dimensions Shown Weighs 9 Lb. 10 Oz. and Exerts a Pull of 2500 Lb. on a Current of 6 Volts and 3 Amp.

same proportion as the ratio of self-energizing. Likewise, should the pressure applied to any pair of brakes show any considerable variation because of lack of proper balancing, the resultant torque effect of the brakes would vary in proportion.

All vehicle brakes used today are self-energized to some degree, and many of them are working extremely close to the critical limit of self-energizing. That is the reason why some of them fail to release.

Buyers of motor-cars are daily becoming more critical in their purchases. They accept, generally without question, the reliability of vehicles sold by established manufacturers. The outstanding points of criticism, therefore, resolved themselves into the obvious elements of car performance which are noticeable in sales demonstrations. Purchasers are primarily critical in regard to acceleration, deceleration and control. This is so true that easy steering has become a sales feature. Similarly, ease of brake control is becoming a sales feature. For this reason, everything possible is being done to reduce pedal pressures and increase the smoothness of brake action on motor-cars.

External Pressure Boosters

Having utilized virtually all of the self-energizing available, engineers have turned to external augmenting devices to assist in the application of brakes. The booster brake is coming into increasing use for this purpose. Fundamentally, the booster is a mechanical substitute for an extremely healthy leg-muscle, as it increases the pressure exerted through the brake pedal.

For satisfactory results, the use of booster equipment must be accompanied by an increase in the rigidity of the brake-operating mechanism, whether it be mechanical or hydraulic. In the hydraulic system, pressures are substantially increased by booster operation, with resulting increase of stresses in tubing, couplings and cylinder units. The possibilities of leakage increase, and the entire installation is raised to a more critical operating point. In a mechanical hook-up, cross-shafts particularly must be given attention so that the angular deflection due to increased torsional forces will not affect a reasonable balance between the four wheels.

Some booster installations have a noticeable time-lag, due to the necessity of exhausting the booster cylinder before the augmenting force which it furnishes becomes noticeable on the deceleration of the car.

Equalization Problems

The problem of vehicle-brake equalization covers a vast territory, and only the more obvious points can be considered in a discussion of this kind.

We might as well start with a tacit admission that there is no such thing as perfect equalization of brakes. Too many variable factors are involved in brake performance to make this possible. From an operating standpoint, brakes can be equalized to such a degree that no inequality is noticeable in road performance, but even under these conditions brakes may be far from actually exerting the same retarding torque on all four wheels. Slight differences in tire inflation and in the condition of tire treads and the irregularities of the road surface are all destructive of equality in brakes.

When it is considered that the brake torque must be transmitted through two friction-media, the brake liner and tire tread, and that these two friction-media are subject to differences in actual friction-characteristics, the possible variation can readily be appreciated.

As the amount of self-energizing of any brake is increased, the difficulties of obtaining close equalization are likewise increased. The ratio between the input and output forces exerted on the brake-band in a full-wrap brake may be as high as 9 to 1. Obviously, this multiplying factor will apply in multiplying the inequalities in the friction characteristics of the brake-lining.

Effect of Variable Friction

Consider two brakes, one having a liner with a coefficient of friction of 0.40, and the other a coefficient of friction of 0.38. The first brake will, for an applied shoe-pressure of 500 lb., exert a retarding force at the drum of 4500 lb., while the second brake, for the same applied force, will exert a retarding force of 3960 lb., a difference of 540 lb.

Whether or not this difference will cause a noticeable drift on the road depends upon the mass of the vehicle and the stability of the front-axle construction. It is very difficult to predict just how far this lack of brake equality can be permitted, because what might be called the general road balance of the car has so much to do with the results obtained.

The differences in performance outlined do not become a factor of any magnitude with brakes having slow action and only moderate power. It is when really fast-acting and powerful brakes are used that these conditions become critical.

The problem of equalization of mechanical brake-linkages is generally understood, and the elimination of friction existing in the linkage mechanism is receiving careful attention from engineers. The effect of torsional distortion of the cross brake-shafts, which adds further difficulties to getting a truly equalized set

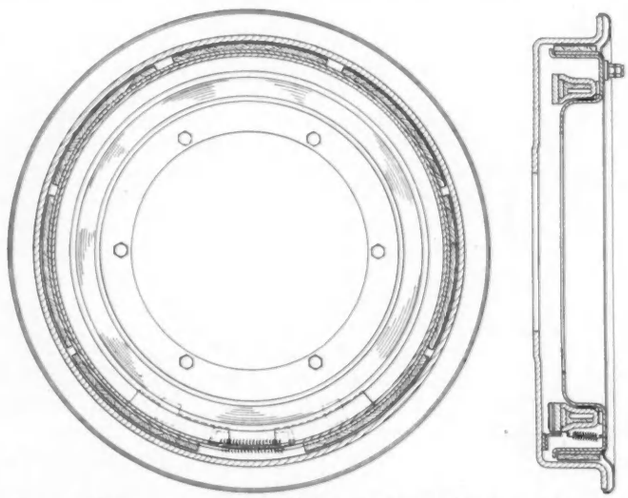

FIG. 2—DRAWING OF 16 X 1¾-IN. ELECTRIC-BRAKE ASSEMBLY

of brakes is being overcome by the use of much larger shafts and rigid mountings.

In the case of hydraulic installations, the problem of getting equalized pressure to all brake-shoes is simplified. This, however, has not completely solved the problem of equalization, because a pressure balance within the hydraulic system cannot be established until all the brake-shoes connected in the system have made contact with the drum.

As considerable self-energizing exists in hydraulic brake-shoes, the shoe that first hits the drum, by reason of self-energizing, can exert a certain amount of braking force before the full-pressure balance has been established within the system. The more self-energization such a shoe has, the more noticeable will be the lack of equalization in the brake performance.

When booster equipment is added to either mechanical or hydraulic-brake units, the equalization problem is increased in the same ratio in which the booster amplifies the brake pressure. Booster equipment, after all, is merely the addition of more pedal pressure; in other words, it is an increase in the forces applied to the brake mechanism. Under these circumstances, equalization problems are increased in the ratio of amplification.

Division of Braking between Axles

Until two years ago a feeling existed among engineers that the maximum amount of braking should take place on the rear wheels.

As considerable transfer of load from rear to front axles occurs during the deceleration of the car, it is obvious that the front wheels have more braking ability than the rears. For example, at 60 m.p.h., a 5000-lb. car stopping at 22 ft. per sec. per sec. will show a transfer of load from rear to front wheels of 655 lb., assuming the center of gravity to be 25 in. above the ground. This gives the two front wheels an additional 490 lb. of retarding ability, assuming a coefficient of adhesion of 0.75 between tire and road. This same load is, of course, removed from the rear wheels.

This load transfer is readily established by the formula:

$$X = \frac{WHd}{32.2 \times L}$$

where

d = deceleration, in feet per second per second
H = height of center of gravity, from the ground, in inches
L = length of wheelbase, in inches
W = weight of vehicle, in pounds
X = transferred load, in pounds

If the desired deceleration is established and the approximate height of the center of gravity of the loaded vehicle is known, this formula can be used to determine the front-rear braking-ratio that can be utilized. It will be noted that the transfer is independent of the initial velocity of the vehicle and varies with the deceleration.

It is now generally acknowledged that at least 60 per cent of the braking effort can be exerted through the front wheels of a car. In fact, actual road-tests have demonstrated that the limit of front-wheel braking effort that can be used is determined by the ability of the front spring-suspension and front-axle assembly to resist brake torque without undue distortion.

Front-end design has not kept pace with the continually increasing brake torque being applied to the front end of cars, and braking effort that will slide the front wheels will reverse the caster on most front axles and make a vehicle difficult to steer during an emergency brake-application. This condition is further aggravated by the fact that not all pairs of front springs will show equal deflection under brake applications, causing a different amount of movement of the two ends of the front axle. Since this movement is resisted by the steering-gear connecting-rod, the apparent effect of the entire combination of circumstances gives the impression that the car wants to turn to one side or the other because of brake inequality. As a matter of fact, the other factors stated have more to do with this lack of stability than have unequal brakes. This condition is particularly noticeable at high speeds, which impose an element of impact loading.

Both the available pedal pressure and the permissible mechanical-advantage ratio between pedal movement and brake-shoe movement are definitely limited. Consideration of the figures noted in the beginning of this paper indicates the reason why engineers are utilizing external augmentation devices to increase the braking ability of their vehicles.

FIG. 3 — NON-ROTATING PORTION OF ELECTRIC BRAKE

FIG. 4—ROTATING PORTION OF ELECTRIC BRAKE

Energizing Brakes Electrically

These and the factor of simplicity of installation led to the development of the electromagnetic brake.

A definite amount of kinetic energy is available in every moving vehicle. This kinetic energy must be absorbed to stop the vehicle. The electromagnetic brake simply utilizes the kinetic energy of the vehicle to stop itself.

Before considering the brake in its commercial form, an indication of what is possible with small wattage consumption in the way of electromagnetic clutches may be interesting. For example, an electromagnet like that shown in Fig. 1, weighing 9 lb. 10 oz., connected to a 6-volt storage-battery and using 3 amp., has a direct lifting ability of 2500 lb. When utilizing a friction face having a coefficient of friction of 0.4, this will render available at the magnetic center-line a torque resistance of 1000 lb.

There is practically no limit to the power that can be built into one of these small electromagnets, and the current consumption need not be excessive.

Consider this force applied to a brake-band designed to utilize the maximum safe amount of self-energizing and you have a brake torque-capacity of 40,000 lb-in. Basically, this is what the electric brake does.

The electric brake, a drawing of which is shown in Fig. 2, consists primarily of an electromagnetic energizing unit and a simple brake-shoe amplifying unit. The electro-magnetic unit comprises a magnet and an armature. The magnet, shown in Fig. 3, is a ring-type unit made from magnetic iron having a U section. Within this U section is contained a coil of copper wire, proportioned for the work required and imbedded in bakelite cement. This magnet is faced with a special friction facing which is also attached to the magnet proper by bakelite cement. The entire assembly is baked under conditions to obtain a rugged and solid unit that is waterproof.

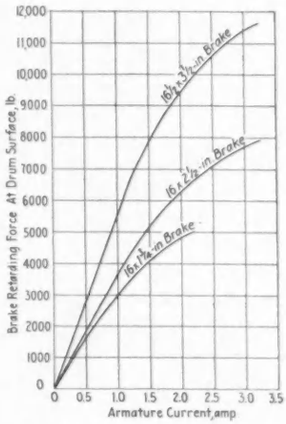

FIG. 5 — PERFORMANCE CURVES OF THREE SIZES OF ELECTRIC BRAKE

The armature unit of the magnet combination is substantially a flat ring of magnetic iron, which is fabricated to withstand the conditions of operation. This armature unit, shown in Fig. 4, is attached to a standard type of brake-drum and revolves with the drum, making very light contact at all times with the friction face of the magnet. The mounting of the armature is by means of alloy-steel springs, which are treated for maximum physical properties. These springs are three in number and evenly spaced around the armature. They are arranged to have a possible longitudinal movement of ⅜ in. so that they will accommodate the position of the armature to that of the brake assembly on the axle and also accommodate the armature position in case of misalignment of the wheel and drum. Thus, if a drum or wheel happened to run considerably out of true, the armature would at all times be in its proper position to function.

The magnet unit has projecting from its periphery two lugs, made from alloy steel, heat-treated and hardened. These lugs are interconnected with the brake-band, which is a steel ring of substantial proportions. Stamped pieces at the ends of the brake-band interconnect with the magnet and also form the abutment against the anchorage of the brake backing-plate, which takes the brake load. Riveted to this band are a number of molded brake-liners, arranged to cover almost the entire periphery of the drum. This form of brake-band gives a very uniform build-up of pressure due to the self-energizing action of the band and accommodates, to a certain degree, eccentricity of the brake-drum.

The action of the brake is extremely simple. When energized by current from the battery, the magnet exerts a gripping action on the armature and rotates with it to a limited extent, as the magnet is mounted on a bearing that is concentric with the brake-drum. This rotation continues until the brake-band expands against the drum surface, when no further rotation of the magnet is possible. From this point, the armature slips on the magnet facing until the current is released, but the drag on the magnet continues, maintaining the pressure on the brake-band and the operation of the brake.

Upon releasing the current, the entire assembly is returned to the neutral or off position by the action of the spring which connects the two ends of the brake-band. The action is the same whether the car is running ahead or in reverse.

Brake Is Nonadjustable

No provision is necessary to adjust the brake-band for wear of the liner, because the magnet unit will follow up the brake-band until contact is made between the liner and the drum surface regardless of the clearance between them. This take-up is limited on passenger-car brakes to ⅛ in. of wear of the brake-lining, after which relining is necessary.

The amount of braking exerted is dependent upon the amount of current passing through the magnet, and this is controlled by the amount of pedal depression. The control of the brakes, therefore, is no different from that of any conventional car, with the exception that the brake-pedal pressure may be just as light as the operator desires. The operator does not furnish any of the pressure necessary to do the actual braking; all that he does is to move a graduated switch.

The curves in Fig. 5 show the amount of brake retarding-force that is obtainable for definite current consumption from brakes of various sizes. Any predetermined amount of braking can be built into this type of brake according to the particular vehicle to be equipped and the rate of stopping desired. This has the advantage that the brake structure cannot be overloaded if the original design is made adequate for the predetermined loads.

The switch or brake controller is a rheostat of unique design. As shown in Fig. 6, it embodies a group of bronze leaves. A curved arm, when moved by the pedal, connects with these leaves successively. The first leaf is connected to the battery, so that the arm carries battery current as soon as the arm connects with this leaf. The next leaf of the switch is connected to the stop-light on the car, and all the following leaves feed into the brake circuit. As each leaf is picked up, a certain amount of resistance is removed from the brake circuit, with a

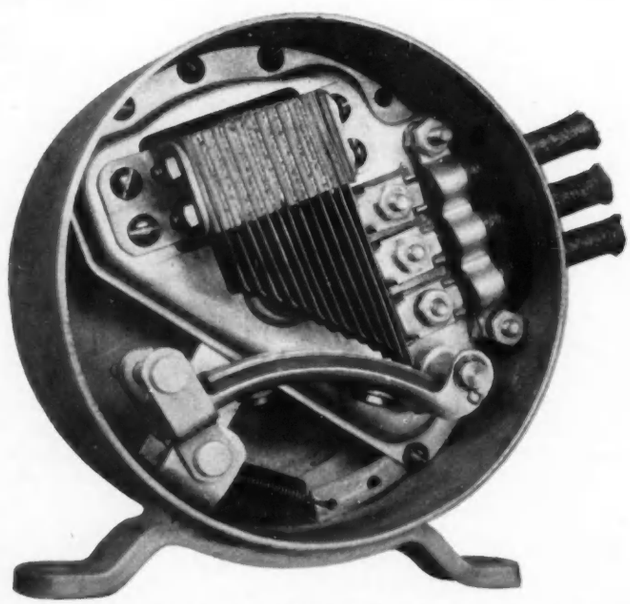

FIG. 6—ELECTRIC-BRAKE CONTROLLER WITH COVER REMOVED

consequent increase in the amount of current flowing to the brakes. All the resistance units in this controller are arranged in parallel, so that, should one or more of these resistance units burn out, it would not affect the operation of the others. Since the last step of the controller is a direct connection to the brakes, complete failure of the brake controller is guarded against.

Overload fuses are included in the brake circuit and are arranged so that, should one brake become grounded for any reason, the corresponding brake, front or rear, will also be cut out of the circuit. In this way either front or rear brakes will remain in operation; but a condition will arise that will be noticeable to the driver, and the fault can be rectified without a complete failure of the brakes.

Since this type of brake is energized electrically, possible variations in value of braking ability are readily obtainable at the will of the driver. A manually operated switch, that can have any desirable number of intensity points, can be installed on the instrument board in any convenient location. The number of points supplied is usually four. This switch can be moved at the will of the driver to give any amount of braking desired. The amount usually is determined by the condition of the pavement, whether icy, wet or dry. The use of this switch does not interfere in any way with the control; it simply creates the same graduation of control at uniformly less intensity than in the full-brake position.

From the standpoint of chassis installation, the electric brake has the advantage of simplicity. The only connection necessary is a single wire to each brake. The controller can be mounted at any convenient point and connected to the brake-pedal by a simple linkage. The wire recommended for brake installation is of very rugged mechanical construction, having, in addition to the usual insulation, a covering of treated loom woven right onto the structure. This wire has been found to be virtually indestructible in brake service. Care must of course be taken to see that the wire is securely fastened at suitable points and that strictly electrical connections are relieved from mechanical stresses.

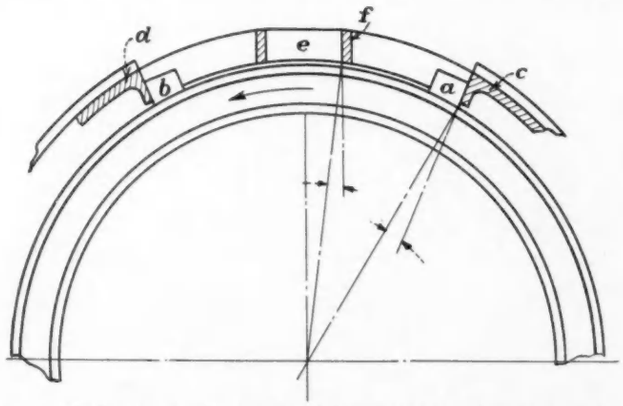

FIG. 7—DIAGRAM SHOWING MECHANICAL OPERATION OF ELECTRIC BRAKE

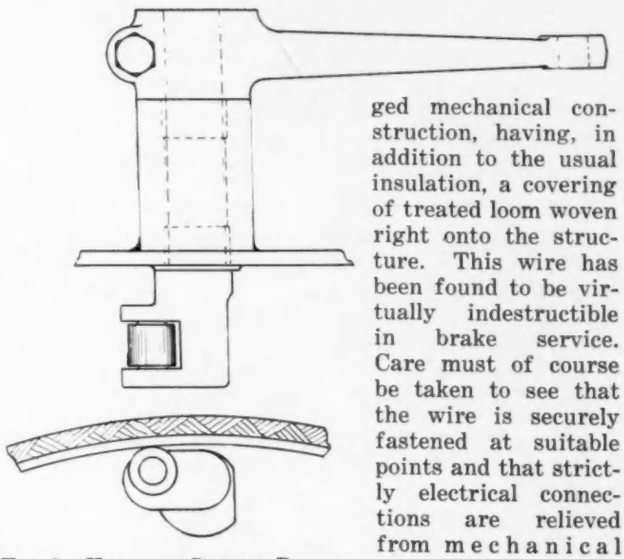

FIG. 8—HOW THE SERVICE BRAKE-BANDS ARE APPLIED BY THE HAND-BRAKE MECHANISM

Automatic Reverse Braking

It is possible with the electric brake to install back of the transmission a friction-operated switch that will be open or off as long as the car is traveling ahead. When, on a slight incline for example, the car starts to roll backward, this switch will close and automatically set the brakes on all four wheels. This unit is disconnected electrically by a switch that opens when the gear-shift lever is shifted into reverse. This method of holding the car on a grade has the advantage that the work is performed by the wheel brakes and not by a special propeller-shaft brake installed specifically for that purpose.

Measuring Brake Performance

In reference to brake performance, it is possible to get very erroneous results in measuring brake performance on cars. When a car is traveling at 88 ft. per sec., small fractions of a second make a wide difference in the apparent stop obtained. In making stopping tests, the use of a pistol, the trigger of which is operated electrically, gives very satisfactory results when properly installed. If the switch for operating this pistol is placed at the start of the pedal movement, the stop measured will be an actual measure of the brake application from the time the operator finds it necessary to set the brakes. Should the pistol switch be placed at the completion of the pedal stroke, a much faster stop will be indicated;

TABLE 1—CAR DECELERATION

Deceleration, Ft. Per Sec. Per Sec.	Initial Speed, M.P.H.																	
	10	15	20	25	30	35	40	45	50	55	60	65	70	75	80	85	90	95
	Stopping Distance, Feet																	
10	10.8	24.2	43.0	63.0	97.0	131	162	218	268	325	385	450	525	606	690	775	871	971
15	7.2	16.1	28.6	42.0	64.5	87.4	108	145	179	207	256	300	350	404	460	516	580	647
20	5.4	12.1	21.5	31.5	48.4	65.5	81	109	134	162	192	225	263	303	346	388	435	485
22	4.9	11.0	19.5	28.6	44.0	59.5	73.8	99	122	148	175	205	239	276	314	352	396	440
24	4.5	10.1	17.9	26.3	40.3	54.6	67.5	90.6	112	135	160	188	219	252	288	323	363	405
26	4.15	9.3	16.5	24.2	37.2	50.4	62.4	83.6	103	125	148	173	202	233	266	298	335	373
28	3.86	8.65	15.3	22.5	34.6	46.8	58.0	77.8	96	116	137	160	188	216	246	277	311	347
30	3.60	8.06	14.3	21.0	32.3	43.6	54.0	72.5	89.4	108	128	150	175	202	230	258	290	324
32.2	3.36	7.50	13.3	19.6	30.0	40.7	50.4	67.6	83.5	101	119	140	163	188	214	241	271	302

in fact, in some cases the apparent deceleration will be amazing. To obtain a reliable comparison of the performance of brakes, it is desirable that the method of testing be carefully standardized, because the error will be very substantial at high speeds if this is not done.

Table 1 shows computed stopping distances for various deceleration rates from different initial speeds.

Preserving Front-End Stability

It is generally conceded that some method other than mere physical force on the part of the driver must be employed to obtain satisfactory brake performance with reasonable pedal pressures. To get the best results that are possible by the use of externally augmented brakes, either booster or power type, means must be evolved to increase the ability of front-end design to resist brake torsional forces. These should include precaution against reversal of caster and provision, independent of the spring suspension, for the absorption of braking forces. The latter is advisable in order to eliminate the irregularities arising from unequal deflection of opposite front springs, and it is particularly important with spring mountings that allow differences to occur in fore and aft position at opposite ends of the front axle.

Another important precaution is to find a neutral position for the steering-arm ball such that the change in front-axle position during brake application will not impose any appreciable load on the steering-gear connecting-rod. If this is not done, the control of the front end of the car will be erratic and give the impression of wandering.

THE DISCUSSION

TEMPLE C. SMITH[2]:—I should be glad to have a more complete explanation of the mechanical action of the brake to show how the motion of the magnet is transmitted to the brake-shoe.

JOHN WHYTE:—Fig. 7 is a purely diagrammatic representation of the brake-actuating parts, most of which can also be seen in Fig. 3. Lugs projecting from the magnet, represented by *a* and *b* in Fig. 7, make contact against inner projecting portions *c* and *d* located near the ends of the brake-band. When the magnet rotates to the left, as indicated by the arrow, the end *d* of the brake-band is moved by the magnet in the same direction. This movement is resisted by the fixed anchorage *e*, against which the end *f* of the brake-band abuts. This causes radial expansion of the brake-band against the drum surface. The surface contacts between *b* and *d*, *f* and *e*, and *a* and *c* are at such an angle from radial lines that they cause definite outward components of the pressure. This results in smooth engagement of the friction liners with the drum surface.

BURTON J. LEMON[3]:—The author presented data based on a coefficient of friction of 0.75; my recollection is that the maximum tire-friction is between 0.72 and 0.80, and that the average is about 0.45 to 0.65. Is there any relation between the coefficient of friction of brake material and that of tires?

Has any measurement been made of the transfer of heat from the brake-drum to the tire of a passenger-car? We know that considerable trouble has been experienced because of this heat transfer on motor-trucks and motorcoaches in cases where the brake-drum is closer than ½ in., and we may encounter the same trouble with passenger-cars if the tire diameter is reduced below 18 in.

An investigation is under way to determine the influence of the space between the rim and the brake-drum on heavy-duty vehicles. This is very important, as the tire manufacturers believe that the tire should not be heated above a limit of 175 to 210 deg. fahr. We know that the temperature does get much higher than that, but it should be kept down to prevent softening of the rubber that holds the plies around the beads.

DUNCAN P. FORBES[4]:—When the wheel diameter is increased, the brake-drum diameter also should be increased to absorb the greater torque.

MR. WHYTE:—Disposing of the heat is the critical problem in all high-duty braking. Brake-drums of passenger-cars are exposed to the airstream under the car, but brakes of motorcoaches, particularly those having dual tires and disc wheels, are very much shrouded, and it is hard to avoid overheating the tires. In one design analyzed recently, it was found better to use 16½ x 3½-in. brakes, rather than 17-in. or 18-in. brakes which seemed to have more capacity, because the smaller brake-drum diameter allowed more clearance between the rim and the brake-drum for ventilation. Radial fins on the drum created a cross stream of air to cool the brake.

There is no relation between the coefficient of friction of the brake lining and that of the tire on the road. The latter varies widely; I have seen cases in which it apparently was greater than 1.

WALTER C. KEYS[5]:—How does the cost of a brake of this type, installed, compare with that of typical mechanical or hydraulic brakes?

MR. WHYTE:—The cost of the electrical brakes, installed in the chassis, does not exceed that of the ordinary mechanical or hydraulic brakes. The difference is that the actuating mechanism of the electrical brake is part of the brake assembly. The only additional installation is the mounting of the controller and its connection to the brakes by means of four wires. We take pains to see that the wires are not called upon for any mechanical functions.

Connection between the controller and the pedal is left to the car engineer, who has his own ideas as to the pedal pressure that is desirable and the pressure can be changed to suit the individual driver by merely changing a spring. It is almost impossible to drive with these brakes without a slight artificial pedal pressure to give a natural feeling in applying the brakes.

C. H. TAYLOR[6]:—Flexible brake-bands cause a tendency toward more brake-lining wear near the heel, where the maximum pressure naturally comes. What provision is made to equalize the wear?

MR. WHYTE:—There is slightly more wear toward the heel end of the band, but the difference is so slight

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[2] M.S.A.E.—Engineer, automotive equipment and labor-saving machinery, American Telephone & Telegraph Co., New York City.

[3] M.S.A.E.—Field engineer, United States Rubber Co., Detroit.

[4] M.S.A.E.—President, Gunite Corp., Rockford, Ill.

[5] M.S.A.E.—Chief engineer, automotive development department, United States Rubber Co., Detroit.

[6] M.S.A.E.—Consulting engineer, Bendix Brake Co., South Bend, Ind.

that the wear is virtually uniform for the first 50,000 miles. When the condition is apparent, the bands can be exchanged from left to right or right to left, and that equalizes the wear. The flexibility of the band allows it to adjust itself to irregularities in liner thickness or in the clearance.

MR. TAYLOR:—What is the clearance between the armature and the magnets?

MR. WHYTE:—There is no clearance; they are in contact.

MR. TAYLOR:—Are they lubricated?

MR. WHYTE:—They are not. The maximum friction drag is about 8 or 10 oz., measured at the surface of the tire.

MR. SMITH:—Is it necessary to provide a hand brake, in addition, on the propeller-shaft?

MR. WHYTE:—Placing a crank and roller, like that shown in Fig. 8, on the horizontal center-line and rotating it by a hand lever creates a direct pressure between the shoe and the drum. The shoe becomes self-energizing, and builds up sufficiently to cause very powerful braking action. This looks as though it would cause distortion of the drum, but no greater radial pressure is required than for other types of application, because only the radial component of the pressure is effective.

J. A. HARVEY[7]:—What is the life of the wearing surface between the armature and the magnets?

MR. WHYTE:—During three years of road work, none of them have been worn out. The pressure between the two surfaces is never higher than about 40 lb. per sq. in., and the duration of pressure of that order would not be more than 2 sec. Brake performance tests show that the greatest desirable brake application for a 3500 lb. car can be obtained with a maximum current of 4 or 5 amp. for four brakes, which also lasts for only about 2 sec., so the current consumption is negligible.

A. W. FREHSE

A. C. BATES[8]:—Only about one-half of the available space in the magnet field seems to be occupied by copper, and the remainder is filled with bakelite. What is the reason for not increasing the amount of copper and thereby decreasing the current required?

MR. WHYTE:—The ratio of copper to iron is worked out to the most economical ratio from a dollars and cents basis. The pressure required largely determines the width of the U section; and this, like every other problem in engineering, is a compromise between ideal operating conditions and cost to manufacture.

MR. BATES:—I raised the question because of the possibility of increasing the life of the contacts in the controller by reducing the current.

MR. WHYTE:—The total current never exceeds 12 amp., and there are 11 contacts measuring about ½ in. in one direction and 1/16 in. in the other. No trouble has ever arisen from this condition.

CHAIRMAN W. R. STRICKLAND[9]:—Some of us who have tried to use full-wrap brake-bands without success have had at least temporary success only by reducing the contact to about 240 deg. Considering the variation in friction coefficients of brake-linings, how can uniform brake action be expected with a full wrap?

MR. WHYTE:—It is important to avoid overloading a flexible expanding brake-band. This band is actually a curved strut. When loaded within safe limits, it operates smoothly; but overloading causes distortion of the band, with resultant grabbing. The amount of wrap that can be used in a brake of this sort is determined largely by the type of service. We can use 330 deg. on heavy commercial vehicles without danger of locking or failure to release. On passenger-car brakes, the wrap can be controlled if necessary by using a liner having a low friction-coefficient of 0.15 for 60 deg. at either end of the band while the rest will have a coefficient of about 0.40. Varying these proportions makes it possible to secure a wrap of whatever amount may be necessary to control the load of the vehicle.

CHAIRMAN STRICKLAND:—Will it be necessary to replace the lining in service with two kinds of material having different friction coefficients?

MR. WHYTE:—Yes.

A. W. FREHSE[10]:—How are the brakes kept in adjustment for equalization in case the radial clearance is 0.15 in. in one brake and 0.30 in. in the brake on the opposite side? What prevents one brake from taking hold ahead of the other, and how are the torques equalized if the arcs of contact are unequal?

MR. WHYTE:—There is no necessity for holding the clearance low; and no observable irregularity has been found because of unequal clearance, although there would be a very slight difference in the actual time of contact.

MR. FREHSE:—A drag of 10 oz. at the periphery of the tire amounts to about 5 lb-ft. We find that this amounts to a difference of about 1½ or 2 miles per gal. in fuel economy, for a vehicle weighing 2500 lb.

MR. WHYTE:—It probably makes some difference but I doubt if the difference would amount to the figure stated. The drag on the brakes for a 2500-lb. job would be less than the figure that I gave, which was for a 3500-lb. car.

M. C. HORINE[11]:—Is it possible to use a rigid molded lining with this flexible internal band?

MR. WHYTE:—We use a hard molded liner, riveted on in sections about 5 in. long, with a space of about ¼ in. between the sections. Even the hardest molded lining has enough flexibility to accommodate the curvature in that length.

R. W. BROWN[12]:—A description of the flexible connection between the magnet and the brake-shoe housing would be of interest, as flexible connections have been a prolific source of difficulty.

MR. WHYTE:—The flexible connection consists of a small spring conduit securely fastened at both ends. The copper conductor passing through this conduit is securely fastened and independent of the conduit. No mechanical stresses are imposed on the electrical connections. The angular movement is small, not exceeding 9 deg.

[7] M.S.A.E.—Operating engineer, Pittsburgh Motor Coach Co., Pittsburgh.

[8] Purdue University, Lafayette, Ind.

[9] M.S.A.E.—Assistant chief engineer, Cadillac Motor Co., Detroit.

[10] M.S.A.E.—Engineer, Chevrolet Motor Co., Detroit.

[11] M.S.A.E.—Sales promotion manager, International Motor Co., Long Island City, N. Y.

[12] M.S.A.E.—Charge of engineering laboratories, Firestone Tire & Rubber Co., Akron, Ohio.

The conduit is made long enough to take this movement with no possibility of fatigue.

MR. KEYS:—When the brakes are applied to stop the car while going up grade, a circumferential movement is caused in the brake-band when the car stops and tends to roll backward. How many inches could the car back, and would the brake-band make a noise when the contact is made in the backing position?

What would be the effect of a broken retractor spring?

MR. WHYTE:—The backlash amounts to ½-¾ in. at the tire. The noise is not noticeable, but it can be heard if anyone listens for it.

A broken retractor spring will cause a brake to drag, but the brake will not lock. The driver would, however, immediately know that something was wrong.

MR. KEYS:—Is any trouble encountered with squeaks in this type of brake?

MR. WHYTE:—Squeaking of brakes is something for which there are as many reasons as there are engineers interested in brakes. Fundamentally, a brake squeak is an audible note caused by amplification of vibrations created by the friction between the brake-drum and the lining. Among the most common reasons for this trouble are liners being loose on the shoe and brake supports that are not stiff enough. Squeaking shows up a weakness at some one of many points in the structure. Nevertheless, every time we encounter squeaking brakes we call upon the representative of the manufacturer of the brake lining.

GEORGE L. SMITH

MR. FREHSE:—Sometimes a squeaky brake can be cured temporarily by merely taking off the drum and putting it back on. The noise is not always due to mechanical weakness, but to the action of the drum, which is rigid at the side where it is mounted on the wheel and flexible at the open side. Applying pressure causes the drum to expand more at the open end than at the closed end. This pressure then causes a lateral component which, coupled with the rotation of the drum, tends to force the shoe out of the drum. These vibrations are lateral and can be measured on a cut-away brake that is set up on a chassis dynamometer. We have observed and measured vibrations ranging all the way from 500 to 1500 oscillations per sec., and these are the ones that cause the worst noise.

F. W. PARKS[13]:—Another squeak cure that sometimes works as well as removing and replacing the brake-drum, as reported by Mr. Frehse, is to blow compressed air into the mechanism.

MR. FREHSE:—We have also found that the powder inside of the brake mechanism has an effect on the noise, and blowing it out will temporarily affect the noise characteristics.

H. F. SCHIPPEL[14]:—In some paper that I have heard within the last few years, a dash adjustment was suggested to vary the relation between the braking at the front and rear according to the road surface. Is there any special provision for adjusting the electrical brakes?

MR. WHYTE:—We use a manually controlled intensity switch on the dash which will provide four degrees of intensity of current for the same pedal action. This switch inserts a graduated resistance in the brake circuit. Changing the ratio of front and rear braking can be done by the use of resistance coils. We find that a car sometimes is inclined to sway on a gravel road if too much of the braking is done at the rear. We have also experimented with a manual control for changing the ratio of the front and rear brakes, but this does not seem to be of enough advantage to justify the complication.

MR. SCHIPPEL:—Are the ratios changed by differences in the magnets or by different resistances in the front and rear circuits?

MR. WHYTE:—Resistance is used in the rear-brake circuit; all the brakes are wired in parallel, not in series.

GEORGE L. SMITH[15]:—A study of the portion of Mr. Whyte's paper that has to do with equalization problems makes it appear that the author is unfamiliar with the method of torque equalization which equalizes the retarding forces of the two wheels at the road regardless of differences in coefficients of friction of the brake-lining and between the tire and the road. This equalization, which I decribed in a paper[16] read at a meeting of the Washington Section, is practically exact up to the point where both wheels are locked. Tests have proved the equalization to be more accurate than that of a differential in equalizing the driving torque.

Mr. Whyte's remarks about the great effect of variation in the coefficient of friction on a self-energized brake are very true, but a pair of torque-equalized brakes can be more easily equalized if the self-energizing action of the brakes is high, as the brake-actuating force is correspondingly small and changes in the distribution of this force are more easily made.

MR. WHYTE:—I am familiar with the method of equalization indicated in Mr. Smith's remarks and have recently had the opportunity to test a car equipped with the type of brake he mentions. The action of the brakes is very uniform and shows remarkable equalization. The prime object of this paper was to show what had been accomplished to date in the application of electrical control to brakes of the vehicle type. It was not the intention to go deeply into the problems of equalization, as that subject merits individual treatment.

[13] Aff. Rep. S.A.E.—Vice-President, Bendix-Cowdrey Brake Tester, Inc., Division of Bendix Aviation Corp., South Bend, Ind.

[14] M.S.A.E.—Design engineer, B. F. Goodrich Co., Akron, Ohio.

[15] M.S.A.E.—Engineer, U. S. Ordnance Co., Washington, D. C.

[16] See THE JOURNAL, June, 1927, p. 754.

Weight and Size Trends in Motor-Truck Development

Annual Meeting Paper

Illustrated with Drawing

By PIERRE SCHON[1]

THE subject assigned to me covers a wide range with such a multiplicity of ramifications that to touch on all various phases as intimated in the descriptive title is impossible. Considering the trend in sizes and weights of motorcoaches first we must frankly admit that the American Electric Railway Association has advanced much farther in seeking a practical code of standards for motorcoach design than has been accomplished in the motor-truck industry, The Uniform Motorcoach Specification is a step in that direction, and in view of the many papers that have been written and presented in the meetings of the A. E. R. A. and the Society in the past, dealing with motorcoach design, I beg to be excused from going into details insofar as the trend of sizes and weights in motorcoach design is concerned.

What has been said about presentations made during the past on motorcoach design also applies to truck design. At the Toronto Transportation Meeting, B. B. Bachman submitted a paper[2] dealing in detail with the problems of the designing engineers and clearly outlining the most important factors that must be observed in designing motor-trucks suitable to meet present and future needs of fleet as well as of single-vehicle operators.

The design of six-wheelers, destined to carry the heavier tonnage loads on balloon tires at passenger-car speed was also most ably discussed during the Toronto meeting in papers submitted by A. M. Wolf[3], L. R. Buckendale[4] and E. W. Templin[5], and more recently in a very interesting paper submitted at the American Petroleum Institute convention in Chicago by C. A. Pierce[6], an authority on six-wheel motor-trucks. However, in leading up to the main subject, we must take into consideration that the design of motor-trucks has gone through most revolutionary changes in the last five years. The vibrating, slow-speed, four-cylinder engines are being replaced with the higher-speed, smoothly operating, six-cylinder powerplants. Crankshaft counterweights, torsion balancers, high-compression heads, better manifolding, better carbureters, high-pressure lubricating systems, oil filters, air-cleaners, fuel pumps, crankcase ventilation, thermostats in the water circulation, better fans and fan belts, larger radiators, more cooling capacity, rubber engine-supports, battery ignition, automatic spark advance, self starters and so on were virtually unknown in motor-trucks three years ago. In isolated cases, some manufacturers had one or more of these improvements at that time, but, when we look back into the models of 1925 and 1926, we realize that the improvements made in truck design in the last three years are more numerous and revolutionary than the combined changes made in the preceding 15 years.

Net payload capacity is the customary basis for rating a motor-truck, but the great variety of actual chassis weights together with the indefinite and uncontrollable range of body weights has resulted in a chaotic condition.

To cure the evil of abnormal overloading the straight-rating formula has been developed, which enables the designer, the manufacturer and the operator to determine the most economical carrying-capacity rating of a chassis.

Lack of uniformity in load-space dimensions is a serious problem from the manufacturers' and operators' viewpoint, 97 different load-space dimensions being available for 35 body sizes ranging from 5½ to 26 ft.

A practical scale of eight load-space dimensions ranging from 39 in. from cab to center line of rear axle for the small 6-ft. delivery body to 135 in. for the 21-ft. moving van is recommended as a solution of the problem.

Chassis Changes

Improvements have been made also in other chassis units and in every detail the 1930 product is better and more reliable than earlier designs, for clutches, transmissions, universal-joints and propeller shafts, steering-gears, chassis lubrication, frames, front and rear axles, demountable wheels and rims have been improved in de-

[1] M.S.A.E.—Sales engineer, General Motors Truck Co., Pontiac, Mich.

[2] See THE JOURNAL, December, 1929, p. 576.

[3] See THE JOURNAL, December, 1929, p. 589.

[4] See THE JOURNAL, December, 1929, p. 583.

[5] See THE JOURNAL, January, 1930, p. 54.

[6] See *American Petroleum Institute Bulletin*, Jan. 2, 1930, section 1, p. 43.

sign, making chronic failures in service a rare exception under normal usage. Four-wheel brakes and air and booster-brake applications also were practically unknown four years ago.

Better steels and materials are being produced and greater responsibility for satisfactory performance of the vehicle is being placed on the designing engineers. The large selection of alloy metals of higher tensile-strength and longer life gives a wider range of activity to the engineer in designing more durable vehicles.

The present tendency of adopting balloon tires on motor-trucks following the almost sudden change from solid tires to pneumatics is bringing about revolutionary changes in the design as well as operating methods of truck chassis. When pneumatic tires were first used on motor-trucks, their application was confined to extreme operating conditions where solid tires would not perform, especially for use in the oil fields. Pneumatic tires on a motor-truck were considered an expensive luxury in those days. The chassis were primarily designed for solid tires and extremely heavy weight in a chassis was supposed to indicate strength and long life. With these slow-moving vehicles, a few thousand pounds' excess weight would not really seriously interfere with the economical operation of the vehicle.

The general adaptation of the pneumatic tire presents a different problem, as excess weight or overloading of a pneumatic tire more directly affects the tire cost per mile than does the overloading of a solid tire. It is also a well-known fact that the actual payload capacity per pound of dead weight is greatly increased where pneumatic tires are used, without endangering the life of the vehicle. This is all brought about by the reduction in vibratory stresses in the load-carrying members of a chassis and the decrease of crystallization, due to the better cushioning provided for in the pneumatic tires.

However, the life of a pneumatic tire can be charted, curves indicating that, as the percentage of overload is increased, the cost per tire mile is also increased. The former method of giving a chassis a nominal payload-capacity rating is not adequate nor practical to fit vehicles built to carry heavier loads at faster speeds on high-pressure pneumatic or balloon tires. Regardless of what has been accomplished by the designing engineers in building vehicles to meet present requirements, the one important question has not yet been solved by our industry, namely, What constitutes an overload?

At the Toronto Transportation Meeting G. B. Ingersoll, of the Federal Motor Truck Co., illustrated this problem and the difficulties to be met in the designing, selling and operating of motor-trucks by stating: "The engineer designs a truck to carry 1 ton, the salesman sells it as a 2-ton truck, and the operator loads it with 3 tons." The customary method of rating a motor-truck by net payload capacity and the large variety of actual chassis weights, together with the indefinite and uncontrollable range of body weights, has been responsible for the chaotic condition existing at present. Figures taken from the *Commercial Car Journal* and presented in Table 1 illustrate the reasons why the carrying capacity of motor-trucks is frequently misunderstood. These figures present minimum and maximum chassis weights in the various payload capacity ranges for chassis with four wheels and with drive through the rear wheels only. By comparing the chassis weights as given, the fact that buyers of motor-trucks, in selecting a truck of a certain carrying capacity, have a wide range in the actual bare-chassis weights is immediately apparent. With these wide variations in bare-chassis weights for the same payload-rating certain vehicles seem to be underrated, while others appear to be overrated for normal service and operating conditions. A truck chassis designed for solid tires, such as many of the extremely heavy chassis listed, may not prove an economical unit to operate on pneumatic tires.

TABLE 1—MINIMUM AND MAXIMUM CHASSIS-WEIGHTS UNDER VARIOUS PAYLOAD-CAPACITY RANGES

Payload, Tons	Chassis Weight Minimum, Lb.	Maximum, Lb.	Variation, Lb.
1/2	1,691	2,290	599
3/4	1,760	2,694	934
1	2,400	4,100	1,700
1¼	2,650	3,774	1,124
1½	2,370	5,300	2,930
1¾	3,000	4,032	1,032
2	3,350	6,200	2,850
2½	4,300	7,500	3,200
3	4,155	8,080	3,925
3½	5,775	10,290	4,415
4	5,100	10,175	5,075
4½	6,208	8,900	2,692
5	6,673	11,800	5,127

TABLE 2—STRAIGHT-RATING EXAMPLE

Front Axle, lb.	3,000
Rear Axle, lb.	7,000
Total Gross, lb.	10,000
Chassis Weight, lb.	4,200
Cab, Body, Special Equipment and Payload, lb.	5,800

State laws on maximum vehicle-weights and maximum weights per wheel or per axle, together with the economic need of reducing the cost of operation, are forcing the designing engineers to strive for a reduction in dead weight without endangering the load-carrying ability or the life of the truck itself.

Body weights, driver control and operating conditions make it virtually impossible to produce a formula which can be universally applied as an answer to the paramount question, When is a truck overloaded? No other branch of the transportation business has tolerated the evil of overloading. Passenger-cars are built to carry from two to seven passengers, and the body design does not lend itself to excessive overloading. In railroading, a car will not be accepted for transit if loaded beyond its maximum capacity as stenciled on the side of the car. The maximum tonnage of steamships has to be registered with the Government, and a boat loaded beyond that capacity is not permitted to leave a port. Only in motor-trucks do we so constantly have to contend with the evil of overloading, although it occasionally occurs in motorcoaches.

Straight-Rating Formula

After many years of abuse, the motor-truck industry has finally located a yardstick that enables the designing engineer, the manufacturer and the operator of motor-trucks to determine the most economical carrying-capacity rating of a truck chassis. This formula or yardstick is the gross or straight rating and is shown in Table 2. This method of measuring the carrying ca-

pacity of a motor-truck has been tried out by several manufacturers and, speaking from actual experience in working the straight-rating policy, I can definitely state that it is a step in advance in curing the evil of abnormal overloading. Using this yardstick of straight rating is a simple matter of elementary arithmetic. The gross rating of a truck chassis starts with the designing engineer, in laying out a coordinated chassis design capable of performing for 100,000 miles or more under a given predetermined gross weight. The designing engineer knows the carrying capacity of front and rear axles not only from laboratory tests but from past performance in actual service. Adding the carrying capacity of the front axle to that of the rear axle gives the total maximum gross-weight for the contemplated design of vehicle of a certain carrying-capacity. Frames, springs and powerplant requirements are figured out to provide ample factors of safety, based on the ultimate carrying-capacity of the two axles not only for carrying a given weight but also to obtain satisfactory performance ability and proper load-distribution.

PIERRE SCHON

Tire manufacturers have established maximum-load figures on tires and to select tires of the correct capacity for each vehicle is a simple matter. One chassis design may be given several gross ratings by using smaller tires than required for the maximum total-gross-weight which, however, does not change the actual carrying-capacity of the chassis. Trucks sold under the straight-rating policy are equipped with tires of ample carrying-capacity, the maximum gross-weight figure applying to the maximum gross-rating of the chassis itself with the largest suitable-tire equipment, while the ratings for the smaller tire-equipment are recommended maximum gross-weight limits. The straight rating solves the over-size tire problem that comes up periodically.

Very often chassis prices are listed in trade publications based on solid-tire equipment, and the custom in the past has been to quote pneumatic tires as an extra. This practice is entirely eliminated with those manufacturers who have adopted the straight rating on their product. In some cases operating conditions permit the use of smaller tires than are recommended by the manufacturer, and when that happens the straight-rating policy is compelling the salesman to warn the buyer that he is taking a risk in abnormal cost per tire-mile. The maximum gross-rating of the chassis itself does not change unless other load-carrying members, such as springs, are changed from the original engineering specifications.

The vehicle is tested out with a considerable overload in excess of the maximum gross figure set for the chassis, these tests being carried out for 50,000 miles or more, to prove that every unit has a sufficient factor of safety to take care of the unavoidable or seasonal overload that the truck may have to carry in certain lines of service. To assume that an overload, over and above the maximum gross straight-rating weight-figure, will cause a chassis to collapse is a mistake. The straight rating does not alter or change the carrying ability of the chassis nor its resistance to abuse, but it definitely provides a yardstick for the engineer to measure all safety factors when designing the chassis, for the manufacturer in selecting the proper tire-equipment and establishing a safe capacity-rating, for the salesman in recommending the right size of chassis to meet certain operating-conditions and, above all, for the operator in selecting the most economical unit to meet his particular needs. A motor-truck which is designed and built to be sold under the straight-rating policy is a coordinated and well-balanced unit, built from the tires up to give good service under a predetermined maximum gross-weight, not underrated nor overrated, but rated to give the best results in the hands of the user. The straight rating is nothing new, for in transportation by rail and water it has been used successfully for many years, so why not for motor-trucks?

The actual payload is not given in Table 2, since it can be determined only after the weights of the cab, body and special equipment have been definitely established. Due to the wide variation in body weights in the various industries, actual payload-capacity clearly depends on the weight of the body, cab and equipment. Variations of 1000 to 3000 lb. between the weights of various body-types suitable for mounting on the same chassis are not unusual, and in some cases excessive body-weight alone has been known to constitute an overload for a certain chassis without having a pound of payload in the body. The straight rating has made considerable progress in the last year and, with a better understanding of its application, it will only be a question of time when the problem of overselling and overloading motor-trucks will be solved, with good results to the entire industry, whether manufacturing, selling or operating.

Standardization of Load Spaces

For many years, manufacturers and operators of motor-trucks have felt the need of standardization in sizes, wheelbases and body load spaces. As far back as 1920, the Society of Automotive Engineers took action on this matter and appointed a committee to investigate the possibilities of setting up recommended standard dimensions for motor-truck bodies. This committee devoted considerable time to this important subject, but, due to the fact that the members were mostly chief engineers from the various truck companies, each one having his own ideas about what the standard dimensions should be, no agreement could be reached. The report of the committee is still in the archives of the Society and it reveals that the committee was working along the lines of standardization of wheelbase dimensions and over-all body-lengths.

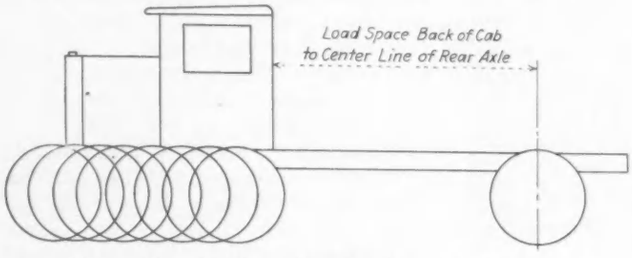
Due to the great variance in front-axle location and the large number of over-all body-lengths, we can easily understand why this committee could not reach any satisfactory agreement. Since 1921, no serious effort has been made by the Society or any other organization

to arrive at a closer standardization in any one of the important fundamental dimensions of motor-trucks. Because of this lack of uniform recommended practice, determining wheelbase and load-space dimensions was left entirely to the individual designing engineers who were again influenced by the demands made upon them from year to year by the sales organizations to increase wheelbase dimensions to be able to mount a body 6 in. or 1 ft. longer than those of competitors. The motor-truck industry has been floundering insofar as uniform dimensions of the product are concerned.

In a recent interview one of the prominent tank manufacturers cited an instance where he built a tank for one of the large oil companies for mounting on a certain truck chassis, shipped the tank to the West Coast and upon its arrival found that this truck manufacturer had changed the load-space dimensions from the cab to the center line of the rear axle by 7 in., although the chassis had the same model designation, thus preventing the proper mounting of the tank on this chassis without costly alterations. This tank manufacturer had used the body builder's prints from his files on that particular model and was not aware that body load-space dimensions had recently been altered.

Standardized Load-Space Will Permit Quantity Body-Production

Body manufacturers throughout the Country are anxious to secure a greater standardization of the most important dimension involved; namely, the distance



Make	Capacity, Tons	Wheelbase, In	Load Space, In
D	1	133	50
GM	1	127¾	52¼
W	2	165	81⅜
GM	2	164	86

FIG. 1—RANGE OF FRONT-AXLE LOCATIONS ON STANDARD TRUCK-CHASSIS

from the cab to the center line of the rear axle. If this dimension could be standardized among the various makes, the business of the body builders could be greatly increased, as certain size bodies would be applicable to any make of truck and could be built up for stock in large quantities. At present, this is impossible, as no one truck manufacturer, with the exception of Ford and Chevrolet, has a sufficient volume on any one model in any one part of the Country to warrant any body builder attempting to build up any one type of body meeting vocational requirements. For instance, if any body manufacturer should attempt to build a department-store body with a load space of 9 ft. in length, 60 in. wide and 60 in. high, he would be forced to design this body for one particular make of truck only. If he should desire to mount this body on another make of truck, he would find that the wheel-housing location would have to be changed. The result of this lack of standardization forces the body manufacturer to confine his large-quantity production facilities to Ford and Chevrolet only. Bodies for other makes must be built on special order unless load space from the cab to the center line of the rear axle happens to coincide with Ford and Chevrolet dimensions.

From the operators' standpoint, this lack of uniformity in load-space dimensions also presents a very serious problem. For a truck body to outlast two or three truck chassis, as in the case with specially designed bodies or tank installations is not unusual. In transferring a body from the old chassis to a new one, the operator may find that the wheelhousing does not fit and the load distribution may be entirely wrong. This not only happens if another make of chassis is used, but actually an operator may purchase the same make of truck of the same wheelbase as the old chassis and find that, due to dimensions being changed in the location of cowl and the length of cab, the load space may vary considerably from the dimensions of the old chassis.

What has been said about body manufacturers' and operators' problems is also of great importance to the truck manufacturers. For the manufacturer to have several models in different capacities taking the same length of body has been a common occurrence; yet, a duplication of inventory was necessary due to the chassis dimensions being different. A 9-ft. body built for a 1-ton truck could not be mounted on a 1½-ton chassis, due to a difference in frame width and also in the dimensions from the cab to the center line of the rear axle. This situation has become so serious that the transportation engineers of several industries have attempted to set up their own standards and are demanding of the truck manufacturer that the product be modified to meet their own particular dimensions. This condition has been brought about by the lack of interest on the part of the manufacturers and the Society who have both failed, until now, to set up any recommended standard practice for motor-truck body load-space dimensions. Manufacturers of motor-trucks today are forced to make alterations in wheelbase dimensions to meet the requirements of certain industries, due to the fact that the manufacturer's standard dimensions are not acceptable. Unless some radical steps are taken to correct this situation, you can readily visualize the difficulties that the industry may have to face in the future. If the major industries should take more energetic action in writing their own specifications, the manufacturing of motor-trucks would soon degenerate into a job-shop proposition; namely, building special wheelbases to meet the standards as set up by the transportation men of the various industries.

Variation in Wheelbase and Load-Space Dimensions

Discussions have been held in the past dealing with the question of standardizing wheelbase dimensions. To obtain results in this direction is impossible, as wheelbase dimensions cannot be standardized, due to the wide range of practical and possible front-axle locations in a motor-truck chassis. The front axle may be located directly below the driver's cab in designs where the cab is placed over the engine, or the axle may be placed in line with the radiator or even ahead of the radiator location, as we find in the conventional types of truck chassis.

Fig. 1 shows the range of possible front-axle locations on standard truck-chassis. No uniformity exists

at present and to seek standarization along this line would be a hopeless task. At the bottom of this illustration, it will be noted that in trucks of equal carrying-capacity and similar design, wheelbase dimensions do not mean anything when compared against the most important one in a motor-truck, load space from cab to center line of rear axle.

During a study of this problem, which started about a year ago, certain facts came to light which made a thorough investigation of the actual body-length requirements for the various industries using motor-trucks, and how the truck manufacturers attempted to meet these demands, advisable. Table 3 shows how the motor-truck manufacturers have been floundering as far as standards in load-space dimensions are concerned and we must admit that it is not surprising that certain important industries find themselves compelled to set up to their own specifications. The first column gives the range of maximum body-length for the standard chassis on six makes as listed in the manufacturers' catalogs. The next six columns give the load space in inches from the cab to the center line of the rear axle. Out of a grand total of 107 load-space dimensions, we find only 10 duplications, leaving 97 diffent load-space dimensions to mount 35 body sizes ranging from 5½ to 26 ft.

In some cases the variations are small, yet they have prevented truck manufacturers, body builders and operators from developing bodies of practical standard dimensions. Where two columns are shown under the same make, changes were made during 1929 from the dimensions as published by those two manufacturers at the start of the year. No criticism of any one of these manufacturers is intended. This table merely shows the actual condition brought about by not having any recommended practice or standard procedure to follow.

In the last two years, most manufacturers have replaced the four-cylinder engine with the more flexible six-cylinder powerplant. In many cases the six-cylinder engine was placed in the same frame, sometimes without lengthening the wheelbase. The hood being longer for the six-cylinder engine, the cab was located several inches farther back on the frame, changing the load space from cab to center line of rear axle. Again we find that cabs have been redesigned to provide for greater driver comfort. In some cases cabs have been lengthened out 3 to 4 in. to allow for more leg room. The important dimension from the back of the cab to the center line of the rear axle was shortened a few inches, as the designing engineer did not relocate the rear axle to maintain previous load-space dimensions.

To forestall any possible cause for criticism in presenting Table 3 containing load-space dimensions of six prominent makes, please note that all of these dimensions are given in trade publications and the manufacturers' catalogs. In addition to this remark, Table 4 shows how our own engineers failed in the past to make it easier for our production department and operators.

TABLE 3—COMPARATIVE LOAD-SPACE DIMENSIONS FOR SIX DIFFERENT MOTOR-TRUCKS[1]

Body Length	Dodge	I. H. C.	G. M. C.	White	Mack	Diamond T
5½	33		33½			
6	31					
6½	39¾	38½				
7						
7½	49¾	50⅝		47	49	
8	50	51½				51½
8½	54¾	51½	52¼	48½		
			58	57¾	58	56 58
9	59¾	55¼	59	62⅝	58¾	59 60
				62⅞		62
9½		61½				
10		65⅞		68⅞	69	64 65 67
10½	69¾	67⅝	70¼	69⅛		70 72
11		73¼	74		74	73 76
11½		81 81½	79	81⅜	77¾	78 79
12		83 81⅞		84⅝	82	81
12½						
13	84¾	85	86		86	
13½			87⅝			89
14				91½	92	91 92 94
14½			97 99		98	96 99
15						102 105
15½	104¾	106⅞		105⅝	103	104
16			111⅝	107⅛	108	109 111
16½						
17						116 113 114
17½						
18			119			121 123
18½						
19 20		131⅞			127 128	131 133
21						
22 23					139 140 145	
24		156⅞			149 152	
25					164	
26					174	
Totals						
35	2 8	5 11	13	11	14 15	28

97 Different Load-Spaces. Grand Total 107.

[1] Body lengths are in feet; all other dimensions in inches.

In one case we find a variation of 1 in. in the wheelhousing location for the same body-length and in another instance a variation of 3¾ in. Another varies 1⅝ in. In another case the variation was 2 in. Thus for 9 maximum body-lengths we find 13 different load-space dimensions from the cab to the center line of the rear axle and 13 different percentages of the load distribution.

Different Frame Widths

What has been said about load-space dimensions from cab to center line of rear axle also applies to width of frame. Body manufacturers and operators are facing the same problem when attempting to build new bodies or transferring a body from an old truck to a new chassis. The wheelhousing may fall in the same place, but, due to the frame being of a different width, relocating the body sills may be necessary.

One manufacturer had standardized on frame-width and load-space dimensions prior to 1930, a step in the right direction. The product of most other manufacturers, however, shows various frame widths, some two and others as many as four different frame-width dimensions. (See Table 5.)

No really good, practical reason exists for truck

TABLE 4—LOAD-SPACE VARIATIONS FOR THE SAME BODY-LENGTH FROM CAB TO CENTER LINE OF REAR AXLE

Model	Body Length, Ft.	Load Space, In.	Proportion of Body Length, Per Cent
T-11	5¾	33½	48.20
T-19 A	8½	52¼	51.23
T-30-42 A	9	58	53.70
T-60 A	9	59	54.62
T-19 B	11	70¼	55.71
T-30-42 B	11	74	58.73
T-60 B	12	79	54.86
T-30-42 C	13	86	57.33
K-102 A	13	87⅝	54.00
T-42 D	15	97	55.74
T-60 C	15	99	55.55
K-102 B	17½	111⅝	53.15
T-60 D	18	119	55.00
Totals	9	13	13

frames showing these variations in width. A thorough search, comparing the needs of all the various industries using motor-trucks, revealed the fact that a 34-in. frame-width is acceptable and the most practical. Only in cases where manufacturers are using the underslung type of rear-spring construction, frames naturally are wider. Outside of this special design-feature in the chassis itself, a 34-in. frame-width meets all conventional needs. Where springs are mounted on top of the rear axle, as in most of our standard American trucks, a truck frame wider than 34 in. is not practical.

Difficulties are encountered when dual pneumatic-tires of the larger diameters are used, due to most of our States having legal restrictions on the over-all vehicle-width. For example, 38 x 9 dual high-pressure tires or 9.75/20 balloon tires have insufficient clearance between tire and rear spring, if the truck frame takes up more than 34 in. of the total width of 96 in. as allowed in most States. This does not apply, however, when rear springs are of the underslung type.

Recommended Standard Load-Spaces

A most careful study of this subject resulted in the findings presented in Table 6. The figures in the first column represent a practical scale of load-space dimen-

TABLE 5—COMPARISON OF MOTOR-TRUCK FRAME-WIDTHS

	G.M.C. 1929		G.M.C. 1930	Other Makes 1929				
Model	Number of Frames	Frame Width Rear	Standard Frame-Width Back of Cab	Dodge	I.H.C.	Mack	White	Diamond T
T-11	1	45⅛						
T-19	2	37			32			
T-30	3	34				33¼		
T-42	4	34	34	34	34		34	34
T-60	4	34					34½	
							36	
						37½		37
K-102	2	38					42½	
Total	16	4	1	1	2	2	4	2

sions from cab to center line of rear axle, ranging from 39 in. for the small delivery body, 6 ft. long, up to 135 in. for the 21-ft. moving van.

In making recommendations for standard load-space dimensions, establishing limits, either plus or minus, would be entirely practical, but these should not exceed 1 in. either way. In a few isolated cases, the manufacturer may have to furnish a chassis to take a longer body than 22 ft. and, to meet those unusual and exceptional requirements, most of the truck factories now have facilities to furnish longer chassis on special order. In view of this fact, this table is confined to normal requirements based on meeting the largest majority of industrial needs.

The entire range of practical body-lengths from 5 to 22 ft. can be mounted on only eight standard load-space dimensions from the cab to the center line of the rear axle and this can be accomplished without running into any danger of abnormal load-distribution. This gives 25 different body-lengths without a greater size-variation than multiples of 6 in. These differences can, therefore, be split again into variations of from 1 to 5 in., making possible the mounting of over 100 different body-lengths without abnormal load-distribution on any one. On the 48-in. load-space, for instance, which is suitable for a 7½-ft. standard body, an 8-ft. body for hauling light bulky materials can be mounted without throwing too much overhang beyond the rear axle. The same 48-in. load-space can also be used for mounting a short body of 7 ft. in length without abnormally disturbing the load distribution. You will also note that even a 6½-ft. body can be mounted on the same chassis without throwing too much weight on the front axle.

TABLE 6—RECOMMENDED STANDARD LOAD-SPACES AND BODY-LENGTHS

Standard Load Space, In.	Shorter than Extra-Short Body, Ft.	Standard Short Body, Ft.	Standard Body-Length, Ft.	Maximum Body-Length Allowed, Ft.
39	5	5½	6	6½
48	6½	7	7½	8
48				
54				
59	8	8½	9	9½
60				
70	9½	10	10½	11
72				
82	11	11½	12	13
84				
99	13	14	15	16
96				
108				
115	16	17	18	19
120				
135	19	20	21	22
132				

A duplication exists in the extra-short and long body-lengths, enabling operators and transportation engineers to select the right chassis with proper load-distribution to suit the requirements of any one of the many industries.

Considerable study has been given this subject by the American Petroleum Institute, the general results of which seem to coincide closely with a study that was made in Michigan based on the broad vision of simplifying the problems of the truck manufacturer, body manufacturer and the operators combined. In other words, these studies were based on taking in the entire nationwide picture of all vocations. In these standard load-spaces, the requirements of any industry can be met and, no matter whether it is a department-store body, a grocery truck, dump body or a 3000-gal. oil tank, you will find that these standard dimensions are suitable and practical for any type of body that is used to a considerable extent.

In the future, should a change in wheelbase dimensions be necessary for some reason or other, absolutely no good reason is apparent why the load-space dimensions from the cab to the center line of the rear axle or the width of frame, if established along practical standards, should ever be subject to alterations. The need for some standardization program is felt throughout the entire industry, and surely the time has arrived when the Standards Committee of the Society should take this matter up and establish a scale of standard dimensions as an S.A.E. Recommended Practice.

THE DISCUSSION

CHAIRMAN B. B. BACHMAN[8]:—The first portion of Mr. Schon's paper deals with the matter of overloading and rating, which is an important item to the manufacturer and to the user. I wonder whether his 10,000-lb. straight-rating is based entirely upon the structural capacity of the axles to carry a burden without thought being given to the tractive ability of the vehicle as evidenced by the size of the powerplant and the gear reduction or whether it is based on tractive ability of the vehicle and the tire capacity to which he has referred. If the former is the case, I think that a vehicle which is rated upon carrying capacity of the axles and tire equipment without reference to the tractive ability as developed by the powerplant and the gear reduction is at some disadvantage in territories as distinctly different as Chicago and Pittsburgh, for instance.

PIERRE SCHON:—The designing engineer takes into consideration the carrying ability of the load-carrying members and the performance ability of the power unit from the engine to the rear-axle gear-ratio to permit that vehicle to operate and give satisfactory performance. Tractive ability is all figured out in designing a vehicle for a given maximum gross-weight. Gear ratio naturally differs depending on operating conditions. When a chassis is used as a tractor for pulling semi or four-wheel trailers, providing a suitable gear ratio to obtain correct tractive ability is essential. Changes through optional equipment to meet various operating-conditions cannot be avoided.

B. J. LEMON

Tire Size and Straight-Rating Method

B. J. LEMON[9]:—Will Mr. Schon tell us some of the drawbacks to the straight-rating method of truck rating, as undoubtedly in his experience and in that of the vehicles which his company has put out some drawbacks have been found? Has he been able to control overloading or is it more of a theory than a fact at this time? Has he experienced any trouble with State legislatures requiring a certain straight-rating in preference to their making use of straight-rating formula?

I think for straight-rating programs the tendency is to put tires on the front axle of the vehicle which will just or a little more than carry the load and put tires on the rear axle that will carry the overload, which is a good feature. If that is the tendency, bigger tires on the rear and not so much excess tire on the front will help the tire companies. That is the one feature which I think is commendable, and I would like to get the general reaction of his company to date on the drawbacks.

MR. SCHON:—The first thing we ran into was complying with legal requirements in those States requiring a payload-capacity rating. The regulations in those States were met by our engineers drawing up a table giving the payload capacity under the old method of rating on each one of our chassis, and this solved that difficulty.

Now, I shall give you a little insight as to what we ran into with our field organization when the straight-rating policy was put into effect. We were actually accused of fighting the tire companies' battles, and in some cases we lost business where buyers insisted on overloading the vehicle. Like everything new, we met some difficulties at the start, but I want to say that these difficulties have been decreasing gradually during 1929, and I do not believe anyone in our organization would recommend that our company go back to the former method of rating our chassis. Our efforts in convincing the operators that it was to their own interest to tire a vehicle properly have met with a very hearty reception in most cases.

The straight-rating does not cure nor eliminate overloading of the tire equipment, but it compels the manufacturer to list a complete selection of various tire-equipment sizes as factory-standard equipment instead of making tire equipment optional. The same size chassis is listed with the smallest suitable-tire equipment and also with the largest and the in-between sizes and each one of them has a different gross-rating. For instance, a 1-ton truck with 30 x 5 single tires all around may have a gross maximum-weight allowance of only 5500 lb., which is sufficient in certain installations. For instance, in department-store delivery

[8] M.S.A.E.—Vice-president of engineering, Autocar Co., Ardmore, Pa.

[9] M.S.A.E.—Field engineer, United States Rubber Co., Detroit.

where loads do not average over 700 lb. in packages and most of the loads are less, 5500 lb. may be sufficient, and the smallest tire-equipment can be used, 30 x 5 or even 6.00 x 20 balloon tires. With the multiplicity of tire sizes suitable for each chassis of a certain carrying-capacity, the range from the lowest recommended-maximum gross-weight up to the highest may be as much as 3000 and 4000 lb., but the tire equipment itself does not change the actual carrying-capacity of the chassis unless the springs are changed. The maximum-carrying-capacity springs are always specified as "standard." Easy-riding springs are provided as a special-equipment item.

J. F. WINCHESTER[10]:—I had not thought of the angle of applying different size tires on a given chassis the way Mr. Schon has just explained it. Much could be learned if, at some future time, someone in his organization could give an analysis of the different factors that constitute the chassis. For instance, we have a chassis that has a very low rating in one case, and we put on a given size tire; with those tires the car carries a small payload. In another case, we put on a larger size tire to carry the heavier load. Will the chassis that is capable of carrying the light load be economical in that work, and when we apply the larger tires with the heavier load, will not the chassis operation be uneconomical because of the fact that we are theoretically overloading the same chassis?

J. F. WINCHESTER

MR. SCHON:—Mr. Winchester, the chassis with the maximum gross-rating figure is not overloaded. The maximum gross-rating in no case takes into consideration the overload under which that chassis has been tested out; in other words, the maximum gross-rating is a safe rating for that chassis. To graduate the size of all the various units in the chassis to meet the weight figures from the minimum figure of the straight rating up to the maximum, with the exception of probably the springs, is impossible. A special spring with thinner leaves and easier riding-qualities can be provided for light loads.

One thing that is required for a manufacturer doing a national business when the straight-rating is adopted is a greater selection of chassis or models than with the former method of rating. No gap must be left in the line, and our company has filled all the gaps which we had last year with additional models of suitable carrying-capacity so that a complete range of maximum gross-weights from 3800 to 28,000 lb. is available. In the tractor-semi-trailer line the maximum gross figures range from 14,000 to 50,000 lb.

MR. WINCHESTER:—Table 1 is very interesting, because it shows a large variation in the chassis weight. For the operators as a whole, it would have been much more interesting if the names of the different vehicles that were evaluated had been given. I bring that point out because in his paper Mr. Schon states that rapid strides have been made in the last few years in the design of motor-vehicles, particularly in the application of pneumatic tires as compared with solid. In the case of the 5-ton vehicle the minimum weight is 6673 lb. and the maximum is 11,800. The latter vehicle could be operated on solid tires and the other on pneumatic tires, as solid tires would probably be uneconomical. For that reason the names in this particular case would have a very important bearing, since two entirely different types of design may be compared in this particular case. Does experience indicate that the heavier chassis is uneconomical on pneumatics?

MR. SCHON:—Mr. Winchester's conclusions are correct. The 6673-lb. chassis is specifically designed to operate on pneumatic tires, with a 5-ton payload. That particular chassis is sold only with pneumatic tires to carry that load, and the heavy chassis of 11,800 lb. is a vehicle designed for solid tires.

Load Distribution

M. C. HORINE[11]:—Some of Mr. Schon's arguments are worth careful consideration, but in some instances I did not exactly see that his conclusions have been proved. For example, the statement is made that the reduction of overloading is one of the benefits of straight-rating. I am unable to see how one particular method of rating affects the practice of the operator or why rating the capacity of the chassis, including body and load, is any different, from the standpoint of the practice of the operator, from rating merely the load alone. As a matter of fact, when Mr. Schon came to that part of the paper in which he announced that a yardstick had been found for the true capacity of the chassis, I thought that he was going to say the pneumatic tire, because I think that is the first definite yardstick that the industry has ever had on the capacity of a motor-truck.

In his analysis of how the straight-rating works, I noticed one very important omission, and that is the effect of load distribution as between the front and the rear tires. Altogether too much of the sort of rating is being practised, wherein the total capacity of the chassis, the gross-weight capacity, is arrived at by adding the rating of the front and the rear tires. That will not give the correct figure. Considering in what manner the gross load is distributed between the front and the rear tires is necessary because otherwise the condition, already mentioned, wherein the rear tires are overloaded as much as the front tires are underloaded, would give the same total, but nevertheless the rear tires will not give satisfactory mileage, and I would earnestly urge in the practice of straight-rating by such companies as are using it that the matter be given very serious consideration.

In answering Mr. Lemon, Mr. Schon said that the rating of a chassis changes as the tire equipment changes. In other words, a chassis having a 4800-lb. straight-rating on a given size of tire would have a greater straight-rating on a larger size of tire. This was a surprising answer to me in view of the impression I gained from the early part of the paper to the effect that the straight-rating was determined by careful engineering analysis and tests and was made on the strength of the structural features of the chassis. If, indeed, the straight-rating depends upon the tire, what

[10] M.S.A.E.—Superintendent of motor-vehicles, Standard Oil Co. of New Jersey, Newark, N. J.

[11] M.S.A.E.—Sales promotion manager, International Motor Co., Long Island City, N. Y.

better or what further rating need we have than the Rubber Association's standard tire-capacity rating?

We have made considerable advance from the old days of flat load-capacity rating. Whether the straight-rating is the final answer remains to be seen, but the fact that it evokes considerable thought on this subject has made it worthwhile, and so I offer one more contribution, which is not of my own origination, nor is it new, but is simply this: From the sales standpoint, the ideal sale of a motor-truck starts, not with the chassis, but with the body. Knowing what the operator has to carry, the next consideration should be what sort of a body should this be carried on. Now what is a truck chassis for? No one operates a truck chassis for any other purpose than to move a body around. If we could move the bodies without chassis, we would do so. When we have determined the weight of the body and its load, then, and not until then, should we consider what size of chassis should go under such a load and what kind of tires are required to support such a chassis and load.

A. J. SCAIFE[12]:—Our company has been using straight-rating for about three years; that is, trying to. It is a hard thing to get over, but we use that as a basis. There is nothing really arbitrary about it. It is an indicator.

For instance, we rate our vehicles, the newer models, as 8000 lb., 10,000 lb. and so forth. The body situation that Mr. Horine has referred to will be taken care of by fitting the different vehicles in between those various ratings. An 8000-lb. chassis, for example, is equipped with single 32 x 6 tires on all wheels. If it is for a dry-goods or a light-weight job and will carry 7000 lb. or less, we use 30 x 5 tires, which are sufficient to carry the load, or, if the operator intends to use the chassis in the oil industry or where the maximum load is over 9000 lb., we put on a 34 x 7, which is the largest tire we can get on that vehicle unless the operator goes out and buys his own wheels. If we know the load to be carried and the weight does not go over that figure, we are willing to stand back of our rating.

I do not think that we use a 70-30 load distribution, such as Mr. Schon shows, even up to 9000 lb. We would carry more of the weight on the front end and get a more equal division of the load on all tires. That is a question of wheel-base with reference to body length.

A. J. SCAIFE

MR. SCHON:—Mr. Scaife has answered some of the questions that Mr. Horine brought up. In regard to load distribution, the question that Mr. Horine brought up, the weight on the front and rear axle, or the distribution of the weight on the two axles, depends to a great extent on the body length and in each case the engineer naturally figures the carrying capacity of the axles so that the front axle is sufficiently strong. The tire equipment always has sufficient carrying-capacity on the front wheels for the shortest body suitable for the chassis; so your 30-70 ratio does not mean that it is standard.The load-distribution ratio may be 35-65, or it may be 40-60.

MR. SCAIFE:—What do those small figures in the first column of Table 6 indicate?

MR. SCHON:—They are the results of the study made by the American Petroleum Institute and are merely put on the table to show how closely the results of the two studies came together.

MR. SCAIFE:—For instance, on the 48-in. load spacing with a 6½ ft. extra-short body, that probably could have a single tire all around, but if you put the 8-ft. body on that same spacing, you would probably put dual tires on the rear.

MR. SCHON:—In both cases the proper tire-equipment would be furnished, singles all around for the extra-short body, while with the longest body we will have practically the entire load space over the rear axle, therefore duals are proper for the rear. For light bulky goods an 8-ft. body can be mounted on that chassis, and it is frequently done. As a matter of fact, you will see trucks on the streets where the load space back of the rear axle exceeds that ahead of the rear axle.

DONALD BLANCHARD[13]:—How do these dimensions with the smaller body compare with those for Ford and Chevrolet trucks as these bodies are produced in large quantities at present?

MR. SCHON:—The dimension of 48 in. from the cab to the rear-axle center-line added to the distance from the dash to the back of the cab equals 100 in. and comes very close to Ford and Chevrolet dimensions. The 1-in. plus-or-minus tolerance will take care of any slight variation. The cabs on the Ford and Chevrolet are shorter than our cabs. A standard panel body, built for Ford and Chevrolet, will go on that 48-in. load space.

I did not completely answer Mr. Scaife's question. The American Petroleum Institute recommendations of 48 in. are exactly the same as the figures derived from the study in Michigan. After months of work out on the Pacific Coast and months of work in Michigan, the figures are identical.

In the next size they have two recommendations, one 54 and the other 60 in., but the latter is only 1 in. longer than the standard figure as obtained from the study in Michigan. In the next two sizes we are 2 in. apart and in the next we are 3 in. In the next to the last size the American Petroleum Institute had two recommendations, 108 and 120 in., as against our 115 in. On the longest load-space, that suitable for a 21-ft. body, the Institute's recommendation was 132 in. and that based on the study which was made in Michigan was 135 in.

In the recommended standard set-up a duplication of body lengths in the extra short and longest allowed is shown. In other words, with eight standard load-spaces any desired load-distribution can be secured. For instance, we have a 6½-ft. body for light, bulky materials. That body is entirely practicable on this chassis, with a load space of 39 in. from cab to center line of rear axle. We also have a 6½-ft. body on the 48-in. load-space. Again we have a duplication between an extra-short body 8 ft. long on the 59-in. load-space and 8 ft. for the longest body on the 48-in. load-space and all the way through. In other words, to cover the same range of body lengths up to 22 ft. in length six manufacturers had 97 different load-spaces. In Table

[12] M.S.A.E.—Consulting field engineer, White Motor Co., Cleveland.

[13] M.S.A.E.—Editor, *Automobile Trade Journal*, Philadelphia.

3, however, we went up to 26 ft. From that number we deduct 12 load spaces, leaving 85 different load-space dimensions, which can be covered with these 8 standard dimensions for bodies up to 22 ft. long.

A. H. GOSSARD[14]:—Operators would, I think, tend to decrease overloading if they would lessen the weight of the body by spending more money for the materials that they use. They hesitate to do that, however, because of the fact that chassis dimensions are not standardized, so that they know that when the body is worn out, it practically has to go into the scrap heap.

American Petroleum Institute's Recommendations

MR. WINCHESTER:—Another group of men, the American Petroleum Institute's Committee on Automotive Transportation, has been working on this subject for a considerable time. In making a study of this subject, the Pacific Group Section considers the following dimensions as vital:

(1) Distance from back of cab to center line of rear axle
(2) Length of frame behind cab
(3) Frame
  (*a*) Over-all width
  (*b*) Height above rear-axle center-line
  (*c*) Slope of frame
(4) Tread
  (*a*) Front
  (*b*) Rear

The recommendations on the establishment of standard dimensions on these items are given below:

TABLE 7—RECOMMENDED DISTANCE FROM BACK OF CAB TO CENTER LINE OF REAR AXLE

Allowable Gross-Weight of Cab, Body and Load, Lb.	Distance from Back of Cab to Center Line of Rear Axle, In.
2,500-4,000	48, 54, 60
4,500-6,000	60, 72, 84, 96
6,500-8,000	72, 84, 96, 108
8,500-10,000	84, 96, 108, 120
Over 10,000	84, 108, 120, 132

(1) *Distance from Back of Cab to Center Line of Rear Axle.*—Establish this dimension in even feet, that is, 48, 60, 72, 84, 96 in. and so on, the range covered by various-sized truck-chassis to be approximately as given in Table 7. In the case of six-wheel trucks these distances are to be measured to a point midway between the two rear axles. In preparing Table 7, the weight of the cab, body and load was used in the belief that this procedure is a clearer basis of rating than the tonnage classification which is far from standardized. The wheelbase would be considered as semi-vital, allowing the manufacturer reasonable leeway in the location of the front axle.

(2) *Length of Frame behind the Cab.*—Establish this dimension with relation to distance from cab to center line of rear axle as given in Table 8. The frame section behind the rear axle should be uniform so that the rear cross-member can readily be moved forward when necessary, to shorten the frame projection. Uniform sections are further desirable where adding rear bumpers on the tank trucks is necessary. Frames tapering in section toward the end present difficulties in both instances. Frame side members should be straight on top, that is, without kickups.

[14] M.S.A.E.—Manager of automotive department, Middle West Utilities Co., Chicago.

TABLE 8—LENGTH OF FRAME BEHIND CAB

Distance from Cab to Center Line of Rear Axle, In.	Overhang behind Rear Axle, In.		Total Length of Frame behind Cab, In.	
	Four Wheels	Six Wheels	Four Wheels	Six Wheels
48	42		90	
54	48		102	
60	48		108	
72	48		120	
84	54	60	138	144
96	60	66	156	162
108	60	72	168	180
120	60	78	180	198
132	60	84	192	216

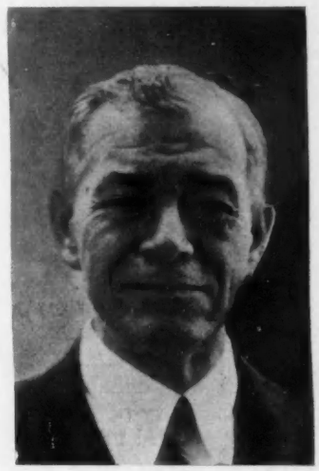
(3) *Frame.*—(*a*) *Over-All Width.*—Frames should be parallel from cab to extreme end, and two over-all widths should be established as standard. An over-all width of 34 in. is now generally used by manufacturers on truck chassis designed for gross weights, cab body, and load, up to approximately 8000 lb. and some manufacturers use the 34-in. width for the entire line. The 34-in. width should be continued for chassis up to a definite carrying-capacity and a second standard width should be established for units above that size, this second width to be 38 or 40 in. The wider frame is an advantage in the larger units, particularly in reducing overhang of flat-bed bodies and providing a wider span for the support of the larger tanks. (*b*) *Height above Rear-Axle Center-Line and* (*c*) *Slope of Frame.*—It would be desirable if the distance between center line of rear axle and top of frame, unladen, and the slope of top of frame with reference to a line drawn through the front and rear wheel-centers could be established within reasonable limits. The measurement between top of frame and center line of rear axle should be as small as possible consistent with proper design of running gear and spring mountings. The frame slope should preferably approach zero when chassis is fully mounted including body of normal weight-allowance but without payload.

(4) *Tread.*—Front and rear treads of single-tired vehicles should not exceed 61 in. On larger vehicles requiring dual rear-tires the tread measurements should be kept to the minimum consistent with turning radius and other features involved in the design of chassis. The possibility of large savings obtainable in having bodies interchangeable between chassis of different makes is recognized universally among fleet operators and need not be amplified.

The comments made in this regard are a review of the situation as submitted by Chairman H. A. McKim and Vice-Chairman W. R. Dand of the 1929 Pacific Coast Group. Needless to say, the majority of other members of this Group feel that the manufacturers should get together.

In my contact with automotive sales organizations, no great stress is laid upon any of these points, and certainly the difference of a few inches in such items as we suggest standardizing would have no great effect upon the sales policy of any individual company. As a matter of fact it might actually promote sales in the motor-truck end of the business.

Self-Maintenance *as Compared with* Service-Station Maintenance



Semi-Annual Meeting Paper
Illustrated with Photographs

By H. V. MIDDLEWORTH[1]

THE AUTHOR says that the conditions under which motor-vehicles operate are so varied that it is extremely difficult to arrive at any fixed or definite rule governing the method of maintaining motor-vehicle fleets, and to determine what the minimum number of vehicles should be to justify the establishment of a self-maintenance organization.

Considering that a large-scale operator is one who operates 40 vehicles or more and that a small-scale operator is one who operates less than 40 vehicles, it is obvious that the problem of the large-scale operator with equipment separated into small fleets in remote localities is in the same class as that of a small-scale operator. With such a condition existing, it seems reasonable to assume that "service-station maintenance" would be more economical considering miles traveled, time out of service and the like, which really should be considered. The successful and economical operation of a fleet of motor-vehicles depends very largely upon the extent of its actual use in the capacity for which it is intended. Let a motor-vehicle cease to function for any cause whatever, and at once it becomes an item of expense. Therefore, any legitimate means which can be employed to eliminate time out of service is a further step toward its successful economical operation.

In summing up the situation, the author says that the vital factor affecting the successful and economical operation of an automobile repair-station is the careful selection of the personnel. If an operator has a sufficient number of vehicles so centralized as to justify the establishment of a self-maintained repair-shop and if he employs a supervisory force equal to that of the manufacturer's service-station, there can be very little doubt that self-maintenance is more desirable than service-station maintenance. If, on the other hand, an operator has not a sufficient number of vehicles so centralized or, if a greater portion of his fleet is operating from several separate and remote districts which would necessitate many miles of travel to and from a central repair-station and an unreasonable period of time out of service, it seems reasonable to consider service-station maintenance the more desirable.

In the prepared discussion following the paper, Martin Schreiber, of the Public Service Coordinated Transport of New Jersey, states the pros and cons of centralization or decentralization of fleet maintenance and F. B. Whittemore, of Mack Trucks, Inc., outlines a satisfactory maintenance plan. F. K. Glynn, of the American Telephone & Telegraph Co., discusses fleet organization and methods, and O. M. Brede, of the General Motors Truck Co., raises various questions in line with whether or not self-service and service-station service should be combined. It is said by H. C. Marble, of the White Motor Co., that low maintenance-costs furnish the incentive for research into the relative economics of both small-scale and large-scale operation.

In the opinion of A. H. Gossard, of the Middle West Utilities Co., who makes comparisons between the two systems of maintenance, it is much easier to provide service and to control it by self-maintenance; for this reason, he believes it will prove the most satisfactory, all things being equal. Criticism is made by Capt. Walter C. Thee, U.S.A., to the effect that there has been lack of progress in maintenance methods. He states that, with the exception of modern shop-equipment installed in present-day service-stations, very little has been done to apply to maintenance of motor-vehicles the fundamental principles or laws of management that have been developed in industry during the last decade; in other words, that the maintenance of motor-vehicles has been stagnating and has progressed not nearly so fast as regards economy and efficiency as has the production of motor-vehicles.

The oral discussion of the paper, as distinguished from the prepared discussion, will be printed in full in a subsequent issue of the S.A.E. JOURNAL.

THE CONDITIONS under which motor-vehicles operate are so varied that it would be extremely difficult to arrive at any fixed or definite rule governing the method of maintaining motor-vehicle fleets. Further, it would be difficult to determine what the minimum number of vehicles should be to justify the establishment of a self-maintenance organization. We speak of large-scale operators and small-scale operators, but just where to draw the line between the two is without doubt a difficult problem. Suppose we call a large-scale operator one who operates 40 vehicles or more and a small-scale operator one who operates less than 40 vehicles. It is obvious that if the large-scale operator has his equipment separated into small fleets

[1] M.S.A.E.—Superintendent of operations, transportation department, Consolidated Gas Co., New York City.

in remote localities, his problem is in the same class as that of a small-scale operator. With such a condition existing, it seems reasonable to assume that service-station maintenance would be more economical considering miles traveled and time out of service, which really should be considered.

The successful and economical operation of a fleet of motor-vehicles depends very largely upon the extent of its actual use in the capacity for which it is intended. Let a motor-vehicle cease to function for any cause whatever, and at once it becomes an item of expense. Therefore, any legitimate means which may be employed to eliminate time out of service is a further step toward its successful and economical operation.

Before any definite conclusion is reached regarding the advisability of service-station maintenance, a careful study should be made of the conditions under which the fleet operates. If it is found that it is broken up into small fleets of say 40 vehicles or less, operating in several separate and remote districts and necessitating many miles of travel to and from a central repair-station, then self-maintenance from an economical point of view seems out of the question. If after careful study it is decided that service-station maintenance is desirable, a thorough survey should be made of the service-station facilities available in the districts from which the vehicles operate. In this connection it is generally understood that service-station managers are glad, and usually eager, to get work from responsible concerns. They are also willing to allow a satisfactory discount on work done for such concerns. It is understood that work of this nature is frequently given precedence over other work that is considered of less importance. Arrangements may be made with service stations to do work at night, thereby eliminating the necessity of laying up the vehicles for repairs during the daytime. Arrangements may be made also to have the painting done over week-ends in order that time out of service may be reduced to the minimum.

FIG. 1—INTERIOR VIEW OF PART OF THE CONSOLIDATED GAS COMPANY'S RECONDITIONING STATION IN NEW YORK CITY

Central-Repair-Shop Methods

Many large-scale operators maintain central repair-stations equipped with modern machinery and appliances necessary to perform the work required in maintaining their fleets. During the last five years the Consolidated Gas Co. of New York has maintained a central reconditioning-station successfully and economically, and I will endeavor to describe briefly its organization, equipment and operation.

Company_____ Div. No._____

Car No._____ Work Order No._____

Date in Shop_____ Date out Shop_____

DESCRIPTION OF WORK

Foreman_____

FIG. 3—FORM USED FOR WORK ORDERS

The first problem was to select a location which would be the most advantageous geographically in order that traveling distances to and from the repair-station might be reduced to the minimum. After several localities had been discussed the most practical location proved to be at 153rd St. and Cromwell Ave., New York City. The repair shop was built on this site and, when completed early in 1925, the plant covered an area of approximately 50,000 sq. ft.

After a year of operation, the plant was found to be inadequate due to the rapid growth of the company's fleets, and an addition of 32,000 sq. ft. was built, making a total area of approximately 82,000 sq. ft. Under this one roof the reconditioning, major repairing and painting of the transportation equipment of the parent company and its affiliated gas and gas-and-electric companies is done. (See Fig. 1.) It is not to be understood that 100 per cent of the work is done by us independently. We have learned that the volume of work in certain specific operations would not justify the installation, maintenance and operation of the necessary expensive machinery and equipment. Such operations as cylinder grinding, chromium and nickel-plating, overhauling magnetos, generators and starting motors, welding and the like, are sent outside to those who make a specialty of such work.

The station is directly in charge of the superintendent of maintenance who cooperates with and is directed by the general superintendent of transportation. It is arranged by divisions, each specializing in the work assigned to it. These divisions are the inspection, dismantling and assembly, engine and transmission, electric truck, electric-truck storage-battery, blacksmith, woodworking and painting, and trimming. A competent foreman is in charge of each division. The work is laid out, supervised and inspected by a general foreman in charge of all divisions, who cooperates with and is directed by the superintendent of maintenance.

When a vehicle enters the reconditioning station for repairs it goes directly to the inspection division where it is thoroughly gone over by a competent diagnostician who indicates upon the inspection sheet shown in Fig. 2

the work to be performed. The inspection sheet then goes to the general superintendent's office for approval before the work card shown in Fig. 3 is made. A card system is kept covering the history of each vehicle. Upon this card are recorded the dates on which each vehicle has been in for repairs, the extent and cost of each repair, and the mileage the vehicle has covered since the last repair. From this card the general superintendent determines whether the vehicle should be repaired, reconditioned, assigned to other service or retired from service, and the result of his decision is recorded upon the inspection sheet.

After the inspection sheet has been approved a work card is made for each division through which the vehicle is to pass, on which is indicated the work to be performed in each division. These work cards go with the vehicle and remain with it until all the work indicated on them has been completed and the cards have been signed by the foremen in charge of the divisions performing the work. The work cards are then attached to the original inspection-sheet and forwarded to the office where the labor and material slips are also attached. By this method it is possible to ascertain the nature of the work performed, the material used and the cost of the labor and material in each division. In fact, the total cost of the job can be computed in a very few minutes after the work has been completed.

By following one of the reconditioning jobs through the various divisions, a better idea may be had of the manner in which the work is handled. The vehicle first goes to the washstand, where it is thoroughly cleaned. This process not only aids the mechanics in their work, but assures cleanness throughout the entire operation of reconditioning the vehicle. The washstand is provided with a double hydraulic lift separated in the center, and is capable of raising any vehicle from the smallest to the largest to the desired height for washing. By this method the washer is able to reach any part of the vehicle advantageously and the process of washing is facilitated thereby. (See Fig. 4.) The vehicle is then sent to the dismantling and assembly division, where it is taken down. The various units are then forwarded to the divisions in which the work is to be performed, such as the engine and transmission, electric-truck, electric-truck storage-battery, and blacksmith divisions. (See Figs. 5, 6, 7 and 8.)

We have standardized our equipment sufficiently to enable us to carry in stock such spare units as engines, transmissions and rear-end constructions. It is therefore no longer necessary to hold the vehicle out of service while the engine and transmission are being overhauled. The spare units are drawn from stock and installed on the vehicle. This process speeds up the work, and enables us to get the vehicle back into service in much less time than would be required by the former method. It also enables the operator to take advantage of making unit repairs when the performance of the vehicle indicates that such repairs are more desirable

AUTOMOBILE RECONDITIONING STATION

Vehicle No. ____ Make ____

Company ____

Speedometer Reading — OUT ____

CONSOLIDATED GAS COMPANY OF NEW YORK
TRANSPORTATION DEPARTMENT
MOTOR VEHICLE INSPECTION REPORT OF REPAIRS

Date — IN ____ Date — OUT ____

Inspector ____

Driver ____

	IN		OUT	
	Ok.	Rp.	Ok.	Rp.
DIVISION No. 4				
Inspection — Vehicle — In				
Inspection — Vehicle — Out				
Testing — Vehicle — In				
Testing — Vehicle — Out				
EQUIPMENT				
Speedometer				
Tools				
Fire Extinguisher				
Mirrorscope				
Curtains				
Skid Chains				
DIVISION No. 5				
LUBRICATION				
Oil Cups				
Grease Cups				
Oil Gauge				
Wiggle				
BRAKES				
Foot Brake Lining				
Foot Brake Adjustment				
Hand Brake Lining				
Hand Brake Adjustment				
COOLING SYSTEM				
Radiator Clean				
Radiator Leak				
Radiator Hose				
Radiator Supports				
Water Pump				
FUEL SYSTEM				
Carburetor				
Throttle Adjustment				
Sediment Drain				
Gasoline Line				
Gasoline Tank				
CLUTCH				
Clutch Action				
Clutch Pedal Adjustment				

	IN		OUT	
	Ok.	Rp.	Ok.	Rp.
Clutch Adjustments				
Universal Joints				
Radius Rods				
Drive Shaft				
Drive Shaft Housing				
Drive Shaft Bearing				
Wheel Adjustment				
Wheel Alignment				
Spark Control				
Gas Control				
TIRES				
Solid Condition				
Pneumatic Condition				
Cushion Condition				
Tire Rims				
Tire Rim Bolts				
Tire Rim Lugs				
Vehicle Cleaning				
Fenders and Running Boards				
DIVISION No. 6				
MOTOR				
Compression				
Cylinders				
Connecting Rods				
Connecting Rod Bearings				
Wrist Pins				
Wrist Pin Bushings				
Pistons				
Piston Rings				
Valves				
Valve Guides				
Push Rods				
Push Rod Guides				
Cam Shaft				
Cam Shaft Bearings				
Timing Gears				
Cylinder Head				
Cylinder Base				
Crank Case				
Motor Supports				

	IN		OUT	
	Ok.	Rp.	Ok.	Rp.
Fan				
Fan Bearings				
Oil Pump				
Motor Timing				
TRANS. & DIFFERENTIAL				
Gears				
Gear Shaft				
Bearings				
Bearing Cups				
Case				
Sprockets				
Chains				
Shifting Levers				
FRONT AXLE & STEERING				
Steering Post				
Steering Post Bearing				
Steering Post Housing				
Steering Wheel				
Steering Worm				
Steering Arm				
Steering Rod Pins				
Steering Rod Bolts				
Knuckle Pins				
Knuckle Bushings				
Knuckle				
REAR AXLE				
Axle Housing				
Axle Shafts				
Axle Shaft Bearings				
Axle Gears				
DIVISION No. 7				
BATTERY				
Battery Connections				
Battery Trays				
Replace Broken Jars				
Replace Broken Covers				
Renew Ceiling Compound				
Equalize Battery				
Charge and Discharge to Determine Remaining Capacity				

	IN		OUT	
	Ok.	Rp.	Ok.	Rp.
DIVISION No. 8				
RUNNING GEAR				
Chassis Frame				
Brackets				
Hangers				
Battery Compartment				
Springs				
Spring Clips				
DIVISION No. 9				
BODY				
Body Sills				
Body Floor				
Body Panels				
Body Roof				
Body Cab				
Body Doors				
Take Body Off				
DIVISION No. 10				
PAINTING				
Burn Off				
Touch-up and Varnish				
Varnish				
DIVISION No. 11				
Elec. work to be done on Gas Cars in Div. No. 1				
ELECTRICAL SYSTEM				
Magneto				
Magneto Connections				

	IN		OUT	
	Ok.	Rp.	Ok.	Rp.
Breaker				
Breaker Points				
Generator				
Generator Regulator				
Starting Motor				
Starting Motor Clutch				
Starting Switch				
LIGHTS				
Light Switch				
Light Fuse				
Horn				
Wiring				
DIVISION No. 12				
UPHOLSTERY				
Upholstery — Clean				
Upholstery — New				
Curtains — Repairs				
Curtains — New				
Cushions — Repairs				
Cushions — New				
Carpet				
Straps				
Roof Cover				

FIG. 2—MOTOR-VEHICLE INSPECTION REPORT OF REPAIRS

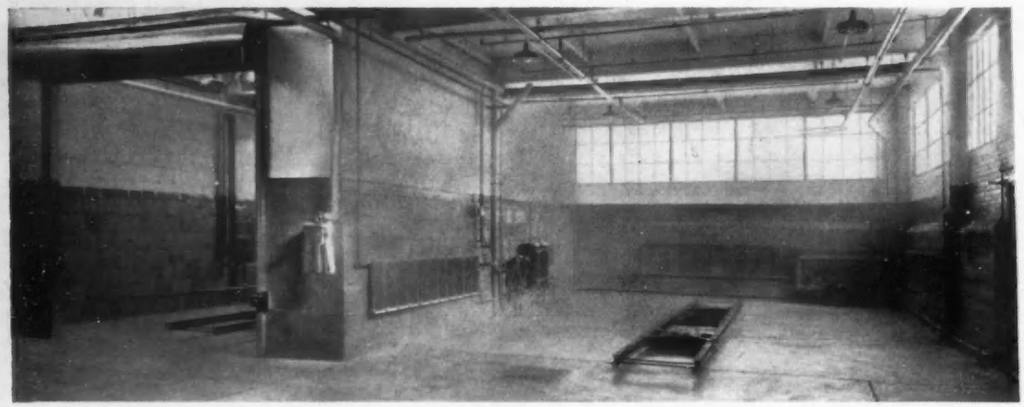


FIG. 4 — SAND - BLAST AND WASHING ROOMS FOR CLEANING VEHICLES

The Double Hydraulic Elevator in the Center of Each Room Should Be Noted. Each Is Capable of Raising Any Vehicle to the Desired Height for Sand Blasting and Washing

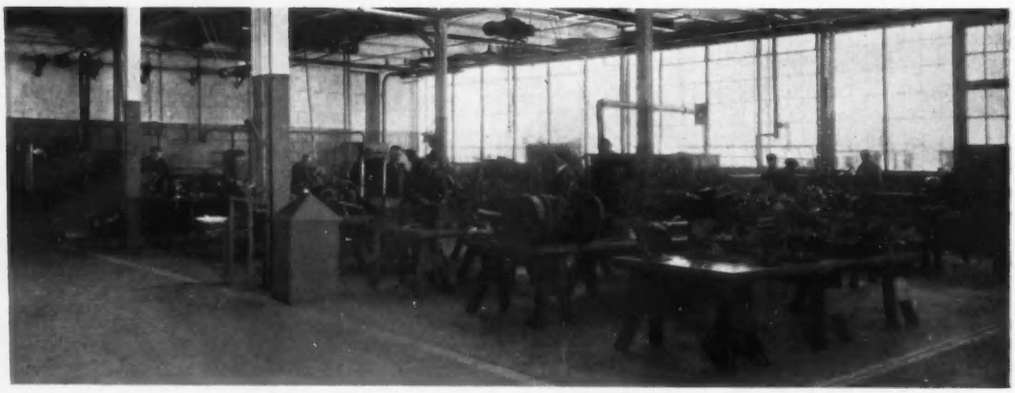


FIG. 5—SHOP USED BY THE ENGINE AND TRANSMISSION DIVISION

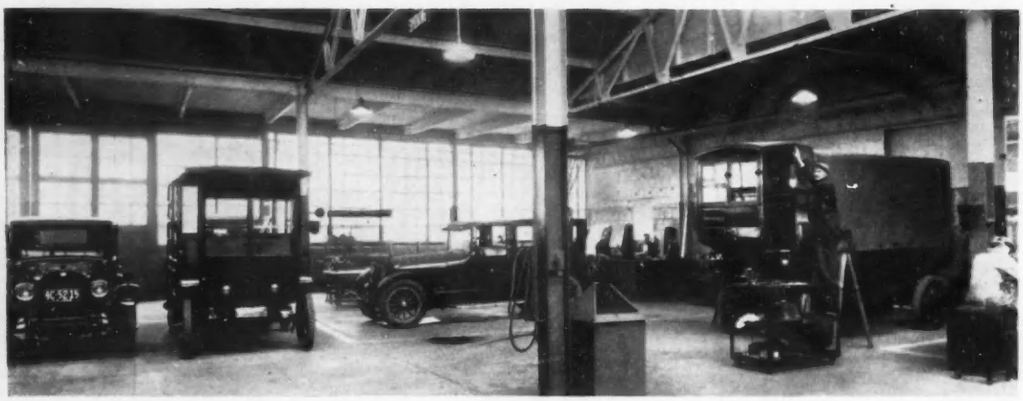


FIG. 6 — ELECTRIC TRUCKS ARE RECONDITIONED IN THIS PORTION OF THE SHOP

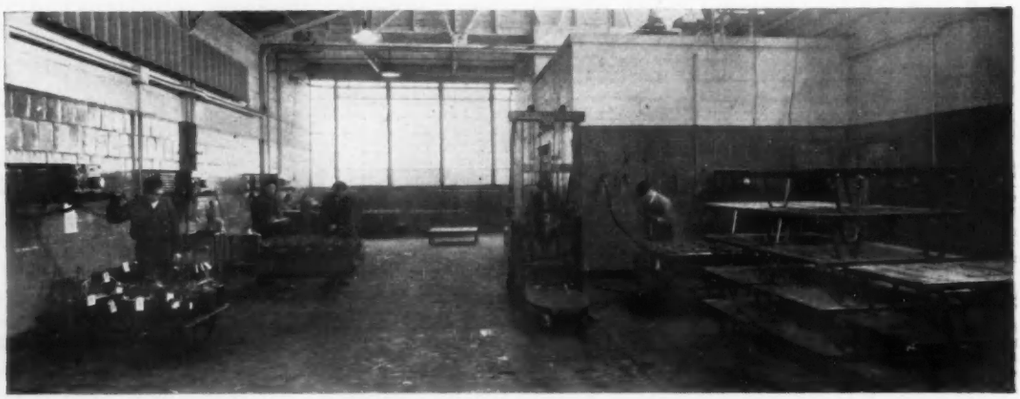


FIG. 7—STORAGE BATTERIES ARE RECONDITIONED AND CHARGED AS INDICATED

FIG. 8—EQUIPMENT PROVIDED FOR BLACKSMITHING AND FORGING

repairs and maintenance of their respective various types of motor-vehicle equipment.

To operate and maintain a service station such as described entails an enormous outlay in property, building, machinery, tools and equipment. It is also necessary to carry a sufficient stock of spare units, repair parts, accessories, materials and supplies. The cost of such an establishment, together with interest on the investment, taxes, insurance, repairs to building, light, heat, power, water and the like, represents an expense which must be accounted for. Does it not seem reasonable to presume that the charges for service-station work must be sufficient to cover the entire cost of maintaining the entire establishment, and is not the customer paying indirectly for this cost? On the other hand, we may ask: Is not the large-scale operator who maintains a central repair-station similarly equipped and operated bearing the same burden of expense? In viewing this question from a number of angles, it might be well to consider some of the outstanding advantages and disadvantages of self-maintenance.

than the reconditioning of the complete vehicle at stated time or mileage intervals. Another method employed in speeding up the work where body repairs are also needed is to send the vehicle to the woodworking division, shown in Fig. 9, and complete the body work first when the other divisions are not ready to start their part of the work.

After all the repair work has been completed, the vehicle is returned to the washstand and again thoroughly cleaned with a chemical preparation mixed with live steam; it then is rinsed with hot water before going into the paint shop. (See Fig. 10.) When all the work has been completed, the vehicle is returned to the inspection division, where it undergoes a final thorough inspection and test before going back into service.

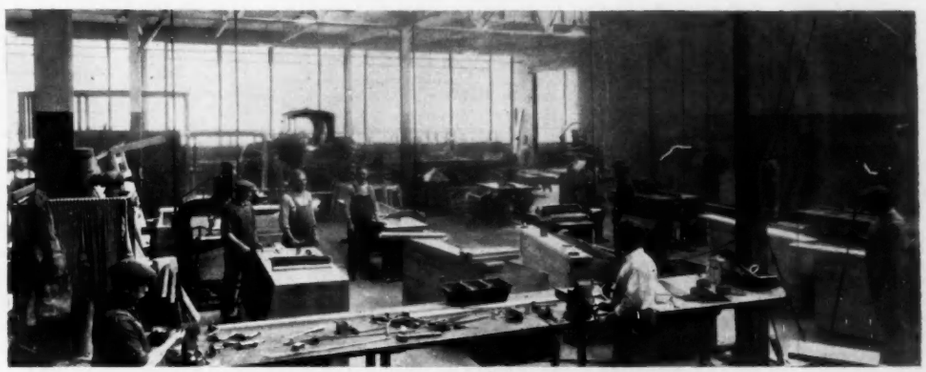


FIG. 9—WOODWORKING AND BODY-REPAIR SECTION

Service-station maintenance with regard to passenger vehicles as well as commercial vehicles is usually found to be satisfactory. Service stations are also well equipped with modern machinery and appliances necessary to turn out a first-class job. They are generally supervised by high-class technical men who are in charge of competent mechanics specially trained in the

First, in a self-maintenance shop the work is constantly under the direct supervision of the person responsible for the maintenance of the fleet. Second, it is not necessary to carry more than a limited supply of spare units, repair parts, accessories, materials and supplies, because they can be procured direct from the service station as occasion may require. Third, self-maintenance repairs might not necessarily be quite so thorough and the vehicle still be perfectly safe to operate, while service-station repairs are generally more

FIG. 10—PAINT-SHOP AND FINISHING DIVISION

extensive. For instance, an operator making his own repairs might make use of slightly worn parts which would render his vehicle entirely safe and in fit condition to operate for considerably more mileage. The service station in most instances, as a matter of policy, would replace a slightly worn part with a new part even though the worn part might still be good for many more miles of use. In support of this statement I refer to a report made by E. C. Wood, of the Northern California Section, which was presented at the meeting of the 1929 Transportation and Maintenance Committee at Detroit in January, 1930, in which it is stated:

> "The requirements of each and every fleet operator are found to differ in many details, such as the amount of work that is maintained in his own shop and the work that is farmed out to commercial shops. In this connection, I wish to bring out that a considerable amount of road delay has resulted from work being farmed out, and in some instances this practice has been discontinued. The reasons for this are, primarily, work not being done properly, not obtaining the maximum life of the part before replacement and the high cost due to early replacement."

In considering some of the disadvantages of self-maintenance, I desire to call attention to a situation which may occur from time to time when the self-maintained repair-shop becomes overburdened. The shop may be working to the limit on jobs already in when, by reason of accident or mechanical failure, a surplus of work may suddenly come in unexpectedly. This condition might require that one or more jobs wait a day or even longer before work could be started on them, which means that the vehicles would be out of service just that much longer. If they were sent to the service station, arrangements might be made to have such work given precedence over other work considered less important, thereby lessening the time out of service.

Maintenance Induces Varied Opinions

Opinions differ widely upon the subject of maintenance. One large-scale operator maintained a well-equipped central repair-station until 1925, when it was discontinued. Since that time all major repairs have been farmed out to service stations. This fleet was scattered over a considerable area, which necessitated many miles of travel to and from the repair station, and this represented time consumed by travel as well as time out of service, both being items worthy of consideration. Some of the savings claimed to have been realized by this change of policy may have been brought about by retiring vehicles on a time or a mileage basis before they have completed their economic life, and replacing them with new vehicles. I have in mind trucks of a certain well-known make which have been retired on a time basis of six years and had covered only from 29,000 to 40,000 miles. This practice is extremely difficult to understand when we consider another large-scale operator who operates 96 trucks of the same well-known make, 49 of them over a period of 11 years, and covering a total of 6,753,670 miles or an average of 137,830 miles per truck; 25 averaging 118,807; 8 averaging 128,209; 12 averaging 167,074; 2 averaging 153,-895; and 2 averaging 222,876 miles per truck over a period of 16 years. All of these trucks are still in operation; they have always been and are still self-maintained. This fleet of trucks is kept in "spic-and-span" condition at all times, and it is claimed by the operator that they are still satisfactorily and economically maintained. This operator was asked how much longer he expected these trucks to operate economically and replied: "As long as I am able to get parts to repair them."

Conclusions

In summing up the situation, it seems to me that the vital factor affecting the successful and economical operation of a motor-vehicle repair-station is the careful selection of the personnel. If an operator has a sufficient number of vehicles so centralized as to justify the establishment of a self-maintained repair-shop and if he employs a supervisory force equal to that of the manufacturer's service-station, there can be very little doubt that self-maintenance is more desirable than service-station maintenance. If, on the other hand, an operator has not a sufficient number of vehicles so centralized or, in other words, if a greater portion of his fleet is operating from several separate and remote districts which would necessitate many miles of travel to and from a central repair-station and an unreasonable period of time out of service, it seems reasonable to consider service-station maintenance.

THE PREPARED DISCUSSION

MARTIN SCHREIBER[2]:—Mr. Middleworth's interesting paper covers a subject of great importance to every motor-vehicle-fleet operator. Each operator must not only determine which maintenance plan to adopt, but also must decide to what extent he will follow the plan selected. There are probably very few operations in which 100 per cent of the work is sent to service stations or done entirely in the fleet owner's shops. Operating conditions are so varied that I agree with Mr. Middleworth that the maintenance plan adopted should be determined by a careful study of each operation rather than by any fixed rule.

An important point has been raised regarding the self-maintenance of motor equipment; that is, is centralized maintenance more desirable than decentralized maintenance? The London General Omnibus Co., which operates about 5500 motorcoaches and which probably is the largest transportation fleet in the world, is an outstanding example of centralized maintenance. That company, however, has a good and sufficient reason for following this maintenance plan aside from the economical aspects. To begin with, it is a police regulation that the equipment must be thoroughly overhauled each year and the territory served has a heavy traffic-density rather than a scattered one. On the other hand, the Public Service Coordinated Transport, which operates the largest fleet of motorcoaches in America, follows the decentralized-maintenance plan.

Decentralized Maintenance Advantages

The principal advantages of decentralized maintenance are: reduction in the time that vehicles remain

[2] M.S.A.E.—General manager in charge of plant, Public Service Coordinated Transport, Newark, N. J.

out of service, elimination of non-revenue mileage to the central repair-shop, and the placing of responsibility where it belongs; that is, provided the local-garage man who is directly responsible is allowed to decide how and when the repairs are to be made. With the decentralized plan in effect, a definite inspection-system of routine adjustments and repairs must be made. This maintenance system should be somewhat flexible, however, as it is found that the most economical maintenance is secured by varying the repair operations to suit the different operating conditions existing.

As each garage is held responsible for its own operating and maintenance results, a keen competitive spirit exists between the various garages. This is, of course, encouraged by the management and, as a consequence, many economies are effected in costs and improvements made in operation. Each garage is practically a research laboratory. Our entire organization is constantly striving to improve results, and is ever on the alert to take advantage of new developments in the automotive field. Public Service Coordinated Transport was probably the first large-scale motorcoach-operator to standardize on four-wheel air-brakes, balloon tires, and the elimination of exhaust heater-pipes in the motorcoach body. Another result of this spirit of progressiveness is the experimental Diesel-engined electric motorcoach that was obtained in Germany last year.

MARTIN SCHREIBER

Decentralized-Maintenance Plan

In the application of a decentralized-maintenance plan certain central operations are still necessary. For instance, a main storeroom, along with a central major-overhaul shop, are required. This main shop is for precise engine and chassis work, and operations requiring considerable tool equipment, such as regrinding cylinder-blocks, boring main-bearings, regrinding crankshafts, electrical work, rebuilding springs, major body-repairs and the like.

The main shop in this case is treated substantially as though it were an outside service-station, and work done for a local garage is carried out under an order from that garage. If, in the opinion of the local garage, the main shop wants to charge too much for a job, the work is either done at the garage, sent to an outside shop, or the operation is changed so as to decrease the cost. As the main shop must therefore compete with outside shops, its costs are constantly being checked and no opportunities are missed to effect economies in operation.

Under a fully centralized system the local-garage manager is compelled to take what he gets, whether the local garage can earn the money to pay for it or not. The garage manager does not know the cost of a job until the work is completed, and cannot therefore be held responsible for the maintenance costs. Ordinarily, under a decentralized plan of maintenance, about three-quarters of the work is done by the local garages and one-quarter at the main shop; but this percentage may be varied according to conditions.

[3] Manager of service promotion, Mack Trucks, Inc., Long Island City, N. Y.

Briefly, the advantages of a decentralized-maintenance plan are:

(1) Elimination of divided responsibility for maintenance
(2) The responsibility for maintenance rests with the person who services and operates the vehicle
(3) That the local-garage man's personal knowledge of each vehicle allows him to effect economies in maintenance not otherwise possible
(4) Time a vehicle is out of service for repairs is reduced
(5) The spirit of competition existing between the various garages results in strenuous efforts to reduce maintenance costs and eliminate road failures
(6) Non-revenue mileage to a central shop is largely eliminated

There are of course numerous examples of successful application of each of the previously mentioned maintenance-systems. However, we have found after considerable experience that not only the most satisfactory operating results, but also minimum maintenance-costs, are obtained with the decentralized self-maintenance plan described.

Influence of Fleet Size

F. B. WHITTEMORE[3]:—Conceding that individual owners and small-fleet operators secure the best, most economical and most satisfactory maintenance from service stations, I affirm that operators of more than 40 units in one district should do their own ordinary running repairs and adjustments and, in some instances, make unit-assembly exchanges. For more pretentious repairs, the choice between central repair-shop and service-station maintenance rests on the convenience and dependability of available service-stations. For general repairs and unit-assembly rebuilding, operators having less than 200 vehicles in one district will serve their best interests by entrusting this work to a factory-branch service-station employing specialized mechanics under superior supervision, having a complete stock of replacement parts, equipment and special time-saving precision-tools; that is, a unit of a service organization operating on the principle that quality parts and quality workmanship at reasonable cost are essential elements of lasting satisfactory service.

Operators must arrange for the garaging of their fleet, in a public garage or in a building either rented or owned. The garage should be located centrally with respect to the routes traversed, unless property values render the cost of the ideal location prohibitive. In some instances, it is advantageous to divide the fleet between two or more smaller buildings in different sections.

Ordinary running-repairs, adjustments, inspections and periodic lubrication call for little or no additional space beyond that required for garage purposes; consequently, they require little additional expense. Besides, this work can be done by the garage crew. The obvious saving of time out of service, and of expense, by not sending vehicles to outside service-stations, coupled with only a slightly increased garage-overhead, indicates that it is advantageous for the operator to do his

own running repairs regardless of the availability of convenient service stations.

Service-Station Availability

The availability of service-station facilities should be investigated carefully before the fleet operator decides on how major repairs are to be accomplished. His investigation should cover the quality of the service-station's workmanship and its ability to turn out work quickly at reasonable cost. He should determine the dependability of the service station, and the interest taken by its management in the performance of his vehicles.

The operator should then count the cost of establishing and maintaining a central repair-shop requiring additional floor space, machines, tools, equipment, a stock of replacement parts and a crew augmented by skilled mechanics and supervisors. Elements entering into the additional cost are taxes, depreciation on plant—or rent, if the building is not owned—depreciation on tools and equipment, interest on investment, repairs to building and equipment, upkeep, power, light, heat and extra labor, supervision and clerical overhead. Such surveys may indicate in some instances that the operator's best interests will be served by relying upon service stations for all general repairs and, in others, by maintaining a central repair-shop.

F. WHITTEMORE

A compromise is sometimes advantageous wherein only ordinary repairs to controls, brake re-lining, carbon and valve work, straightening and removing dents from sheet-metal work, the renewal of ignition wires, spark-plugs, hose connections and water-pump parts and the removal and replacement of unit assemblies such as engines, transmission, rear axles, radiators, magnetos and carbureters are handled by the operator, major repairs being turned over to service stations. While his expense and overhead naturally will be greater than that required for adjustments and running repairs only, the greater convenience often justifies it.

Convenience must be taken into consideration when the time required for a repair or for the exchange of a unit assembly is less than a working day, whereupon the time consumed in taking the vehicle to and from the service station becomes relatively important. The central-repair-shop operator should not neglect to include the cost of carrying an inventory increased by the value of the parts and unit assemblies needed for satisfactory maintenance of this kind. These are the principal points which the fleet operator should consider before deciding either on a central repair-shop or for service-station maintenance. For the type of repairs so far discussed, it is evident that the choice may be governed by local conditions. For major repairs requiring more than a working day I believe that, except in cases of the largest fleets, service-station maintenance is the best.

Factory-Branch Service-Stations

The advantage lies with the good factory-branch service-station when major repairs are to be done and engines or other unit assemblies are to be rebuilt. The operator wants quality workmanship and parts and maximum vehicle availability, all at minimum cost. Factory-branch service-stations can provide:

(1) Quality workmanship, because they have the expensive special tools that are essential for precision work, mechanics specially experienced in their use, supervision and management backed by the cumulative experience of the national branch-organization whose policy is based on the motto "performance counts."

(2) Quick work, because of their many special time-saving tools and fixtures and because their mechanics and supervisors are familiar with the particular work. They have on hand complete stocks of replacement parts, in the absence of which delays are often unavoidable.

(3) Dependable work at minimum charges, because it has been the general policy of motor-truck and motorcoach manufacturers to have their branch service-stations charge for labor at a rate barely sufficient to cover cost plus overhead based on a continuous flow of work. A definite profit is derived from the sale of replacement parts, but that profit is the same whether the parts are sold over the counter or installed by service-station labor. For major repairs on which time-saving precision-tools are used, service-station cost is lower because the average central repair-shop does not have sufficient volume of work to show an adequate return on any considerable investment in special tools and equipment. If time-saving tools are not bought, the number of hours required for repairs is increased and uniform accuracy is more difficult to attain. If the tools are bought for only occasional use, fixed overhead is increased thereby and, the repair operations still take more time than when they are done by service-station mechanics who specialize in certain classes of work.

(4) Generally lessened overhead on labor, because a greater volume of work flowing more continuously through the shop reduces the amount of idle or lost time. Work coming from many sources is less apt to show violent seasonal fluctuations. Likewise, the varying needs of customers enable many service stations to employ night shifts, thus reducing materially the fixed overhead per hour of labor. A constant check on time taken for various operations is possible in a large branch-service-station organization and a flat rate or incentive system reduces operation time and cost without affecting the quality of workmanship so long as piecework or flat rate to the mechanic is not tolerated.

(5) The minimizing of peak loads by calling upon other stations within their organization. This interdependence and exchange of experience on stocking parts, repair methods and special equipment tends to increase the efficiency of each service-station and to make each better able to give quality service, which is the only kind of service that is permanently satisfactory.

Two Plans for a 500-Vehicle Fleet

F. K. GLYNN[4]:—Mr. Middleworth certainly deserves our sincere thanks for his excellent paper. In his introduction he divided the sheep from the goats; that is, the large-scale from the small-scale operators. For the sake of argument let us say that so far as the major problem of self-maintenance versus service-station maintenance is concerned, there is no difference or dividing line because service-station maintenance—and I am talking from experience gained in the motor-truck field alone—should be the more economical for a one or a 1000-vehicle fleet. This last is a bald statement and should provoke arguments pro and con beyond these that I shall give.

Are the companies for which we fleet operators work in the automobile business, or are they selling dry goods, coal, gasoline, public-utility service and the like? Just think about this question. Are our companies specialists along motor-vehicle lines? Has any one of us as many vehicles in his fleet of one particular manufacture in one locality as would be serviced regularly by a progressive dealer for that manufacturer? "Probably not," is the answer. Can we, therefore, afford the relatively large per-vehicle investment in shop buildings, in machinery, special tools, spare parts, spare units and the like? My answer to this is: try and do it from the cold dollars-and-cents standpoint. It is difficult!

Does a fleet operator with an extensive shop "miss seeing the woods because of the trees"; that is, from being all wrapped up in making his shop pay and thereby neglecting the opportunities for economies in the every-day operation of his vehicles on the road? Unfortunately, this condition seems somewhat too prevalent. In this connection Table 1 shows a typical organization for a self-maintained 500-vehicle fleet based upon an average for a number of truck fleets and gathered into one set of figures.

The superintendent of transportation has an office manager, stenographers and clerks. He has an operating supervisor who will require about 5 inspectors, 30 garage mechanics, 10 washers and 10 utility men. To operate the shop, he would have a shop supervisor, about 5 foremen, 30 mechanics, and some stock men. Incidentally, here is a worry: "Overhead on the Stock." Many a fleet has in excess of $100,000 lying on the shelves, unfortunately involving carrying charges up to say 15 per cent annually. The superintendent of transportation cannot do all his engineering work, so he has an engineer. The engineer needs a draftsman and perhaps a clerk to do statistical work. The total motor-vehicle personnel is 106; total operating and shop employes, 85; vehicles per employe, a total slightly under 5; and vehicles per operating and shop employe slightly under 6. With this organization set-up there seems to be a tendency to forget all about preventive maintenance.

[4] M.S.A.E.—Engineer in charge of operation and maintenance of automotive equipment, American Telephone & Telegraph Co., New York City.

TABLE 1—TYPICAL ORGANIZATION FOR A 500-VEHICLE FLEET WITH COMPANY-OPERATED REPAIR-SHOPS, UNDER PLAN 1

Superintendent of Transportation

1 Office Manager	1 Operating Supervisor	1 Shop Supervisor	1 Engineer
2 Stenographers 2 Clerks	5 Inspectors	5 Foremen	1 Draftsman 1 Clerk
	30 Mechanics 10 Washers 10 Utility men	30 Garage mechanics 5 Stock men	

Total Motor-Vehicle Personnel	106
Total Operating and Shop Employes	85
Total Vehicles per Employe	5—
Vehicles per Operating and Shop Employe	6—

TABLE 2—TYPICAL ORGANIZATION FOR A 500-VEHICLE FLEET USING MANUFACTURERS' AND COMMERCIAL REPAIR-SHOPS, UNDER PLAN 2

Superintendent of Transportation

1 Office Manager	1 Operating Supervisor	1 Supplies	1 Engineer
2 Stenographers 2 Clerks	10 Inspector Repairmen 10 Garage Mechanics 10 Washers 10 Utility Men	1 Clerk	1 Draftsman 1 Clerk

Total Motor-Vehicle Personnel	52
Total Operating Employes	40
Total Vehicles per Employe	10—
Vehicles per Operating Employe	13—

Table 2 shows a typical organization for a 500-vehicle fleet using manufacturers' and commercial repair-shops.

F. K. GLYNN

This organization set-up reacts successfully to the practice of preventive maintenance by placing the control of mechanical condition in the hands of responsible men whom we shall call, for want of a better term, inspector-repairmen. The office manager, engineer, and their staffs, remain the same. The operating supervisor has 10 inspector-repairmen, 10 mechanics, 10 washers, 10 utility men, one supplies man, and a clerk. An inspector-repairman has a multitude of responsibilities and duties and these have been fully covered in the paper I represented before the Detroit Section and which was printed in the S.A.E. JOURNAL, August, 1929, p. 148.

Equipment of Service Stations

Are service stations equipped to care for fleet-operators' repairs? Experience indicates that the answer is "yes," and perhaps they are equipped better than for fleet operators now maintaining extensive shops.

Do service stations know how to handle fleet repairs? Let us put a big question mark after this and dwell upon it somewhat.

Until fairly recently, fleet-operator repair-shops were an absolute necessity, because the commercial and manufacturers' repair-shops were of value only as parts-supply centers. Manufacturers, however, realiz-

ing that their interests extended beyond the consummation of sales, brought about a gradual improvement in service-station operation, and this has resulted in a very definite trend in eliminating fleet-operated motor-vehicle shops. The change from company shops to outside shops as illustrated by Tables 1 and 2 decreases the number employed in the motor-vehicle personnel, and the usual plan followed is to employ a small running-maintenance night-crew. Mr. Whittemore covered this feature very satisfactorily.

The first step in the direction of service-station maintenance is preferably handled by inspector-repairmen. Their job is to investigate and approve outside repair-shops at strategic points in a territory. This preliminary approval of an outside shop, based upon reputation, equipment and the like is, of course, contingent upon future satisfactory work performed at a reasonable cost. Naturally, rather close observation and supervision on the part of the inspector-repairmen is necessary until the commercial shop is fully educated to the requirements of the particular fleet. When repair work is assigned to commercial repair-shops, the amount of work to be done, the parts to be replaced, the quality of the workmanship, the time out of service and the total cost, are the responsibility of the inspector-repairmen. Further, all bills for repairs should be carefully scrutinized by the particular inspector-repairman before being passed to final approval for payment.

Fundamental Aim of the Fleet Operator

The fundamental aim of a fleet operator lies in keeping the vehicles on the road 100 per cent of the working hours, keeping them safe and satisfactory to operate and doing so economically. This, experience indicates, can be accomplished with the cooperation of the commercial repair-shops. General suggestions along these lines are to

(1) Promote competition between commercial repair-shops both as to price and workmanship.

(2) Educate thoroughly the commercial repair-shops as to the needs and requirements of the particular fleet.

(3) Inspect a certain percentage of the repairs in progress and finished, seeing that only those repairs and parts necessary to give satisfactory operation are provided.

(4) Urge commercial repair-shops so far as is practicable to carry in stock rebuilt spare-units for the fleet vehicles.

O. M. BREDE

As Mr. Whittemore says, tinkering jobs belong in the fleet garage at night. This does not as a rule require much in the way of special equipment, and naturally one cannot afford to send this small work out of one's own garage. Where a unit goes bad or gets beyond the point of tinkering repairs, the commercial repair-shop should be utilized. If spare units are carried by the commercial shop, it is possible to install them there at night and to make repairs to the defective units during the regular working hours, thus safeguarding against time out of service. In closing, let me present what seems to be the present conception of the job of a superintendent of transportation.

Transportation Superintendent's Duties

(1) Build up an efficient operating organization; incidentally, the number of men the superintendent supervises should be no gage of the importance of his job.

(2) Determine the most efficient type, make and size of chassis, body, accessories and auxiliary equipment for the job.

(3) Determine the economic time for replacement.

(4) Arrange for the purchase of all automotive equipment, maintain an adequate stock of parts and supplies, and arrange for the disposal of replaced equipment.

(5) Determine the practices to be followed in the daily maintenance and in the repair of the vehicles.

(6) Locate, lay out, and equip garages and shops.

(7) Collect, compile and study operating data to effect improvements and economies.

(8) Measure operating results.

(9) Study the use of the vehicles in the field with a view toward increased productivity and the elimination of lost time.

(10) Determine the qualifications of the drivers and train them in the safe operation and efficient care of the vehicles.

(11) Review proposed motor-vehicle laws.

(12) Handle registrations and licenses.

Combining Both Systems Suggested

O. M. BREDE[5]:—Having given considerable thought to the subject, I believe that the correct answer to the problem of self-maintenance as compared with service-station maintenance is a combination of both systems.

We must necessarily keep in mind that the problem involves small, medium-size and large fleets, centralized and decentralized fleets, fleets standardized on one or two makes of vehicle, and fleets in which are many makes of vehicle. Some operators are now equipped for and operating virtually 10 per cent self-maintenance, others are partially equipped and operating at partial self-maintenance, and a third group not so equipped is using service-station maintenance. Any attempt to set up certain rules by which to classify operators definitely and in such manner determine for or against either self-maintenance or service-station maintenance would, I am sure, fail to accomplish that which is desired.

To determine the most economical system of maintenance an individual analysis should be made of each fleet, taking all factors into consideration. If this analysis is conducted carefully by some one person or persons thoroughly trained in the operation and maintenance of motor transport, the best system for that particular operator will be found.

Condensing all of the various types of fleet operations mentioned into three general groups, I believe they can be termed: (*a*) Small fleets, which include large-scale operators with widely decentralized fleets, such as an

[5] M.S.A.E.—Director of service, General Motors Truck Co., Pontiac, Mich.

operator with 400 vehicles operating in 50 cities; (*b*) medium-size fleets; and (*c*) large fleets.

The small-fleet operator cannot justify the cost of equipment necessary for self-maintenance, and this class of operator is most economically served by service-station maintenance.

The medium-size-fleet operator can, we believe, operate to advantage by adopting a combination of both systems; that is, operating his own inspection, washing, greasing and running repairs, and taking advantage of the facilities of the service station for major repairs.

The large-scale operator, because of his ability to spread maintenance-equipment costs over a large number of vehicles, and because of more frequent use of such equipment, can better justify the expense of setting up for self-maintenance. To this class of operator the manufacturers' service-stations can be of considerable help, by offering to assist the operators' shop mechanics, foremen, garage superintendents or superintendents of motor equipment, in a better knowledge of the vehicles and the most efficient methods of performing the various maintenance operations, also making known the availability of various labor-saving tools and devices, thus assisting the operator in the matter of keeping a well-balanced parts-supply and thereby preventing so far as possible parts shortage, surplus and obsolescence. This type of cooperative contact is certain to increase the operator's shop-efficiency, decrease his maintenance cost and minimize the time that vehicles are out of service. Arrangements also can be made by the large-fleet operator who is equipped for self-maintenance to send a portion of his work to the service station when his own shop is carrying a peak load.

H. C. MARBLE

Manufacturers' Service Stations

The manufacturers' service-stations offer three distinct services to the fleet operator; complete maintenance, partial maintenance and cooperative contact. They are equipped with every modern device, having constantly in mind the shortest and best method of performing each separate operation. The personnel is educated and directed by the factory, and new methods are quickly adopted nationally.

The majority, if not all vehicle manufacturers, have a technical field-force that travels throughout the Country continuously. These men analyze trouble-reports, and these are reviewed by the factory service-division for the benefit of the engineering department's advices in the matter. Each situation is speedily reduced to fact and, when proved to be sound, is at once transmitted to all field service-stations. When they are not sound and do not get to the seat of the trouble, the findings are speedily rejected. Thus some of the "ouija-board" service which does occur is certainly less likely to occur in service stations which are factory controlled or factory directed.

Vehicle manufacturers have specialists available for the type of analysis and cooperative contact referred to, for the definite purpose of assisting operators in the solution of maintenance problems. Naturally, the manufacturer is desirous of increasing his volume of service-station work, but I feel that the three systems of maintenance I have mentioned all have their proper places, and that the decision to adopt any one system or combination can be reached only by a careful individual analysis of each fleet operation.

Low Costs the Mutual Incentive

H. C. MARBLE[6]:—A truly dispassionate discussion of Mr. Middleworth's paper by anyone actively and vitally interested in maintenance problems is about as difficult to secure as an unbiased judgment of a baby show from a mother whose own offspring is one of the contestants. However, for the purpose of this discussion, private prejudices and personal pet theories have been relegated to the ash heap and, if they appear later, it is without conscious intent. Nevertheless, in spite of this high resolve, conclusions reached are bound to be shaded by personal experience, and service policies later quoted should be considered as applying more directly to the organization with which I am connected than to the industry as a whole.

Looking at the entire maintenance problem, it is not inconceivable to believe that, in the year 2030 or thereabouts, some as yet unborn magician of automotive engineering may pull the rabbit from his hat in the form of a motor-vehicle which will function perfectly without attention for its allotted period of years, and then disintegrate as a unit to be finally disposed of by simply shoveling the debris off the road and onto the junk pile. Thus at one time will be solved all problems of parts, repairs, and trade-in of used cars. Awaiting the vague and probably distant arrival of this millennium we are forced to consider the maintenance problems as they actually exist, and not as they should be.

In considering the matter of maintenance it is basically important to recognize that the fleet operator and the manufacturer, though traveling by different roads, arrive at the same destination of identical interest through a common desire for low maintenance-costs. The fleet operator naturally desires the economical operation of his fleet, for obvious reasons. The manufacturer, without seeking to claim that he has constituted himself a charitable welfare-committee for distressed motorists, finds that he must take an active interest in low maintenance-costs if he desires to continue the sale of his product.

Whether the operation of maintenance facilities is profitable or not, these facilities are a necessity and represent an integral part of the sales program of the manufacturer. Therefore, since it is to the business interest of the manufacturer, from the standpoint of future sales, to endeavor continually to lower maintenance-costs, we find his desires and those of the fleet operator to be as closely linked together in the matter of efficient and economical maintenance as are Siamese twins.

Specifically considering Mr. Middleworth's paper, it seems that a discussion of it can well be treated in the manner of a track meet in which there are a number of

[6] A.S.A.E.—General service manager, The White Motor Co., Cleveland.

conceded or uncontested events, as there are only a few points on which there could be much divergence of opinion. It seems obvious that the small-scale operator or the large-scale operator with the widely scattered fleet had better employ the facilities of the service station than to attempt self-maintenance. It is equally obvious that the operator of a large and concentrated fleet can install profitably a system calling for some degree of self-maintenance; but the question as to whether or not this system should embrace every variety of repair is open to some argument.

Large Concentrated Fleet Considered

Considering only the operation of the large concentrated fleet, it is readily granted that the operator can apply self-maintenance very efficiently in the matter of scheduled periodic inspections and immediate running repairs to the items requiring attention as brought to light by these inspections. However, all mechanical products by their very nature at some time or other require a complete overhaul and, when the operator comes to the point that seems to indicate the necessity for a thorough going over of the chassis and a tearing down and rebuilding of the major units, there may be some sound variance of opinion as to the most profitable procedure. Today, many operators are finding it to their advantage to use previously repaired major units commonly called "replacement units," rather than to repair the major unit removed from the chassis and await the completion of the job for reinstallation in the same chassis. Such replacement, of course, effects a great saving in time.

A. H. GOSSARD

In using replacement major units the operator need not be deterred from making this time-saving through fear of having a high cost for the replacement unit, for the cost is based upon the cost of bringing the unit removed up to standard, rather than upon a fixed cost per unit. If it is found that the policy of using unit replacements does cost slightly more, it also usually will be found that this extra cost is more than made up by the lessened time the transportation unit is out of service. In favor of the service station, it can be said that they are fully equipped to handle these major-unit repairs more efficiently because their volume of such business justifies their having the necessary mechanical aids, a personnel trained specifically for this business, a complete stock of repair parts, and the entire operation under direct factory supervision.

Factors Governing Parts Replacements

It has been suggested that it is often the policy of the service station to replace parts in cases where the operator, under self-maintenance, would be able to get still further use from them. Here it may be said that, with the exception of replacement-unit overhauls, the efficient operator of a service station will base parts-replacements upon the desire of the customer, as low maintenance-costs are to his interest as much as they are to the customer's. Cases exist in which following the desires of the customer would lead to dangerous and probably expensive operation, here the function of the service station is much like that of a physician of preventive medicine offering the benefit of experience and expert advice on maintenance ills.

From the foregoing it should not be gathered that any claim is made that there is any great mystery about the operation of a service station, or that it requires any particular ability that is not at the command of the fleet operator who employs competent help. The question of whether the fleet operator can or cannot save money by self-maintenance is largely a question of volume of major repairs. If his volume of such business justifies the securing of equipment, space and able personnel, he probably can handle his own repairs for less money, as it is only logical to assume that the service station hopes at least to operate at some profit.

Justification of Capital Investment

One other point must be given very serious and grave consideration by the fleet operator before he embarks on a program of complete self-maintenance. This point revolves around the justification of the capital investment required for the establishment of complete maintenance-facilities. Consider, then, the fictitious case of an operator who has, by commendable industry, built up the largest wholesale dairy business in his particular city. For the purpose of expediting the delivery of his products to the ultimate consumer he has, over a period, acquired a large fleet of trucks and is operating them in a relatively concentrated area. His business has prospered. He has made and is making money in the dairy business.

Armed with a variety of facts and figures, the operator interviews his banker for the purpose of negotiating a loan to install complete self-maintenance facilities. The operator tells the banker that, after a careful study of the entire maintenance question in connection with his trucks for a period of three years, during which time he kept an accurate record of costs, he finds that he can save a total of $10,000 per year by adopting a program of complete self-maintenance. The banker asks how much it will cost the operator in the form of capital investment for space, equipment and so forth. The operator replies that the amount required will be in the neighborhood of $100,000, on which there will be very little depreciation.

Then the banker asks the operator whether he has considered that, by investing $100,000 directly in the dairy business, the operator can, in accordance with his past record, make $20,000 yearly instead of saving $10,000 yearly, telling him also that the operator is in and knows more about the dairy business as a whole than he does about the automotive-maintenance business. In conclusion, the banker advises the operator to reconsider this maintenance saving and proceed about the employment of his capital in the spot where it will bring the operator the greater return.

The foregoing case undoubtedly is extreme, as few operators would go in for complete self-maintenance at one step, but would, instead, build up to it over a period of time. The underlying principle as to capital investment, though, is the same whether it is put in at one time or over a period.

The question of capital investment in automotive-

TABLE 3—MIDDLE WEST UTILITIES CO. AUTOMOTIVE COSTS; YEAR, 1929

Co.	No. of Garages	No. of Cars	No. of Makes	Total Miles	Total Cost	Cost Per Mile	No. of Trucks	No. of Makes	Weight, Tons	Total Miles	Total Cost	Cost Per Mile	Cost per Ton-Mile
WP	8	133	16	1,780,670	$92,972.07	$0.052	175	16	192.5	1,424,031	$112,240.41	$0.079	$0.070
CI	3	142	24	2,526,291	135,154.38	0.054	408	16	546.5	3,026,304	278,619.27	0.092	0.070
KU	1	57	9	733,744	45,040.05	0.061	100	11	128.5	857,632	85,683.61	0.100	0.067
PS	5	25	8	339,032	24,324.44	0.072	153	11	170.0	1,509,677	109,187.60	0.073	0.066
SG	2	55	11	692,241	54,413.02	0.079	91	8	109.0	699,184	69,005.75	0.098	0.082
CP	3	87	13	1,503,757	87,638.65	0.058	197	10	233.0	1,820,553	157,850.49	0.086	0.073
IN	1	16	6	247,508	10,960.67	0.044	54	7	62.0	504,911	37,926.57	0.075	0.065
KE	3	35	10	380,868	26,763.83	0.070	32	11	33.0	287,208	23,425.87	0.082	0.079
Total	26	550	..	8,204,111	477,267.11	*0.058*	1,210	..	1,474.5	10,129,500	873,939.57	0.0865	*0.071*

maintenance facilities is one that every fleet operator who embarks upon a system of complete self-maintenance must consider seriously. Presumably, he originally selected his own business because he could make the most money at it and, unless overburdened with investment capital, he should give some time to finding out whether the initial expense of maintenance facilities actually will mean a concrete saving effected or the loss of a further possible profit.

In addition, all businesses have their ups and downs, and the operator who is at one time crowding his fleet may at another time find a fair part of it idle. When the days of necessary retrenchment are at hand, the operator can dismiss his excess maintenance-personnel in short order; but he cannot so easily rid himself of the expense of carrying his permanent maintenance-equipment through times of depression.

In conclusion, it can be said fairly that the operator who is endeavoring to decide between self-maintenance and service-station maintenance should first determine by careful investigation that self-maintenance will or will not bring to him a dollars-and-cents saving. If his research shows him that such a saving actually can be made, he should ascertain whether the amount of the saving is a return upon the capital investment required that is commensurate with the return which would accrue to him through the employment of a like amount of capital more directly in the line of his own business.

1929 Maintenance-Cost Comparisons

A. H. GOSSARD[7]:—The subject discussed in Mr. Middleworth's paper is of interest to all fleet operators, and of especial interest to utility operators. The subsidiaries of the Middle West Utilities Co. operate in 30 of the 48 States, all these States being east of the Rocky Mountain States. Each subsidiary company is regarded as a self-contained operating-unit under a decentralized plan of operation, with the executive head held responsible for results. Operation of automotive equipment comes under whatever plan appears to be the most feasible for obtaining the best results. Many subsidiaries have no company garages, while others service all automotive equipment with company garages.

It should be possible for us to determine whether there is any advantage in maintenance by one of these methods, or by the other. Therefore, we will compare the operating costs of automotive equipment in a number of the subsidiary companies that are operating garages with those of subsidiary companies that have no garages.

The automotive costs for the year 1929 for some companies using a garage system for self-maintenance are shown in Table 3; those of some companies using service-station maintenance are shown in Table 4. These figures do not show that companies using service-station maintenance entirely obtain any better results than companies having one or more company garages. The company showing the lowest operating-cost maintains the equipment by a company garage. The equipment, though, is limited to a few makes and is operated on a high load-factor and over a long period. Close supervision makes this possible.

TABLE 5—ACCIDENT RECORDS OF MIDDLE WEST UTILITIES CO. FOR AUTOMOBILE OPERATION; YEAR, 1929

Company	Equipment: Passenger-Cars	Equipment: Motor Trucks	Equipment: Total	Total Accident Cost	Accident Cost per Vehicle
CI	133	406	539	$9,671.82	$17.94
CPC	10	25	35	238.80	6.82
CP	86	210	296	9,326.68	31.51
KE	29	34	63	367.08	5.83
KU	57	95	152	3,752.91	24.68
NP	56	19	75	306.48	4.09
PS	99	79	178	3,126.12	17.56
SG	43	105	148	2,524.07	17.05
WT	159	69	228	3,679.08	16.14
WP	122	194	316	8,551.02	27.06

Generally, it is our accounting practice to conform to the practice of automotive dealers in servicing equipment. Charges are made for repairs to equipment on a basis that includes the overhead of conducting company garages in the labor charge. For example, the labor charge, if costing 75 cents per hr., is charged for at the rate of $1.50 per hr. It is not our practice, though, to have company garages show a profit. If the doubling of the labor charge at the end of the year shows a credit to the garage, an adjustment is made in the labor charges. The difference between the cost of labor and the charge for labor is carried in a suspense account, and balanced with indirect charges to the garage.

[7] M.S.A.E.—Manager of automotive and transportation department, Middle West Utilities Co., Chicago.

TABLE 4—MIDDLE WEST UTILITIES CO. AUTOMOTIVE COSTS; YEAR, 1929

Co.	No. of Garages	No. of Cars	No. of Makes	Total Miles	Total Cost	Cost Per Mile	No. of Trucks	No. of Makes	Weight, Tons	Total Miles	Total Cost	Cost Per Mile	Cost per Ton-Mile
WT	0	62	10	1,133,000	$63,996.91	$0.057	186	10	192	2,573,133	$184,500.61	$0.072	$0.07
NP	0	54	14	624,989	36,149.56	0.058	21	6	26	174,582	18,839.69	0.108	0.087
CPC	0	11	8	139,676	7,836.04	0.056	25	7	22.5	271,620	18,430.89	0.068	0.075
Total	0	127	..	1,897,665	107,982.51	*0.057*	232	..	240.5	3,019,335	221,771.19	0.0735	*0.071*

On account of the control of service, self-maintenance is the most satisfactory. Either self-maintenance or service-station maintenance calls for supervision. Too much or too little service can come from either method. It is much easier to supervise and control self-maintenance and, for this reason, we believe it will prove the most satisfactory, all things being equal.

Consideration Given to Safety

Safety is important and must be taken into consideration. (See Table 5.) An examination of the accident records of these companies for the year 1929 discloses that those companies that depend on service-station maintenance have just as low, or a lower, accident cost per vehicle as those companies using self-maintenance. Rigid inspection of vehicles at stated periods and drastic rules in case of accidents are having more to do with reducing accident cost than either of the above types of maintenance.

CAPT. WALTER C. THEE[*]:—I desire to state that, after hearing various papers read on maintenance of motor transportation by various members of the Society who are in charge of the maintenance of large motor-vehicle fleets or factory service-stations, the systems employed are practically identical and the fundamental principles similar to those of the average service-station of 10 to 15 years ago. With the exception of modern shop-equipment installed in the present-day service-stations, very little improvement has been made on the system of maintenance.

To be specific, I should say that very little has been done to apply to maintenance of motor-vehicles fundamental principles or laws of management that have been developed in industry during the last 10 to 15 years. I feel safe in saying, and I believe that many will agree with me, that maintenance of motor-vehicles has been stagnating and has progressed not nearly so fast in regard to economy and efficiency as has the production of motor-vehicles. The cause of this stagnation can be explained by saying that mass production has not been applied to the repair of unserviceable unit-assemblies, and economic factors applying to mass production have not been considered in the repair of motor-vehicles.

CAPT. W. C. THEE

Fundamental Principles Summarized

In his Production Meeting paper published in the S. A. E. JOURNAL, December, 1928, p. 568, John Younger showed how efficiency and low production-costs have been attained in the production of motor-vehicles by following certain fundamental principles of manufacturing management based upon economic laws.

(1) Concentration upon a single product
(2) Keeping each article distinct
(3) The extensive use of conveyor systems
(4) Subdividing work so that each worker has only one or very few operations to perform
(5) Providing the required quantity of material of the specified quality at the required time and place
(6) Assigning a definite amount of work to each worker to be done in a given time
(7) Wage payment on the straight day-rate
(8) Acquirement of sources of raw materials
(9) Reduction of inventories of materials in stock and in progress
(10) Foregoing the taking of intermediate profits on processes between raw material and finished product
(11) Keeping materials and parts in rapid motion to assure quick turnover
(12) Using machine tools that give the lowest production cost and require the least manual control
(13) Employment of machines that perform several operations simultaneously with the same amount of labor as for one operation
(14) Sending machines to the overhaul shops at standardized periods
(15) Recognition that the obsolescence factor is more potent than depreciation of machines by wear

Application of Industrial Principles

The application of some of these laws or Fundamental Principles of Industrial Organization and Management to the Maintenance of Motor Transportation is described in detail and illustrated in the paper I presented at the Transportation Meeting of the Society in Chicago, Oct. 25, 1927; published in the S.A.E. JOURNAL for November, 1927, p. 539-547.

I have heard managers of large service-stations say that the unit-replacement system cannot be applied in their organization. It is my belief that it can be applied in either a small or large service-station, as well as by fleet operators, if sufficient thought is given to the subject and each case is analyzed sufficiently so that a satisfactory solution can be worked out.

The unit-replacement system of maintenance of motor transportation has been used exclusively by the United States Army and, whenever properly applied, has been not only very successful, but very efficient and economical. The unit-replacement system is not being accepted by many of the large factory service-stations or repair shops of large-scale fleet-operators for the following reasons:

(1) Managers in charge of maintenance of motor-vehicle fleets or large service-stations are not willing to accept and use the system.
(2) It has not proved to be more economical. This is due to the fact that unit assemblies have not been rebuilt economically in mass quantities.
(3) The equipment of large-scale fleet-operators is not standardized.
(4) Failure of manufacturers of motor-vehicles to cooperate with fleet operators and factory service-stations in regard to making arrangements to repair unserviceable units.

The adoption of the unit-replacement system must be initiated by the manufacturer, for the reason that unit assemblies cannot be rebuilt economically unless the repairs are made to the defective parts in mass quantities.

[*] S.M.S.A.E.—Commanding Officer, U. S. Army, Quartermaster Corps, Second Corps Area Motor Repair Shops, Fort Hancock, N. J.

Whether the units are to be rebuilt and distributed by the factory or by various factory service-stations is a question that will need to be answered by making a study and analysis of each particular case to determine the most economical solution, and whether sufficient quantities of unserviceable unit-assemblies of the same make and model can be supplied by the territory that is considered. In only a very few cases will it be economical for a fleet operator to rebuild his own units unless he pools or merges with other operators, because sufficient quantities of similar unserviceable units could not be supplied to warrant the expenditure of funds for special high-speed equipment and to employ at least one man to perform only one or a very few operations. This man must be supplied with enough work to keep him busy continuously.

Ideal Solution Suggested

Perhaps the ideal solution for the adoption of the unit-replacement system by fleet operators is not to buy vehicles complete, but to purchase units and build the vehicles with them. Col. Edgar S. Stayer, Q.M.C., Commanding Officer, Camp Holabird Quartermaster Depot, in his paper on Standardization of Military Motor Transportation, published in the March-April, 1930, issue of the *Quartermaster Review*, stated that:

> The standardization upon a special type and make of vehicle, so far as military motor-transportation is concerned, is not considered as the proper method of standardization, but rather a standardization based on written specifications describing the units which will be used to build transportation, and which are known by service and testing experience to be of sufficient durability and ability for the military needs.

Operation of this type of equipment will simplify repairs by the unit-replacement system. The following is also quoted from Colonel Stayer's paper as being of interest to operators of commercial motor-vehicles:

> In simplification of cargo transportation for the military service there appear to be three salient factors: (*a*) the number of different vehicles that is required: (*b*) the types of capacities that will be adopted; and (*c*) the interchangeability of units among the different types of vehicle.

A discussion of each of these factors can be found in the foregoing references. It is absolutely necessary that all repairs to unserviceable units be made on a large scale, or the economic factors applying to mass production will be lost and maintenance cost will be increased. I will agree with any discussion pertaining to the desirability of having managers of service stations repair their own units. Whenever this is done, however, the cost of repairing each unit will be much greater, as skilled workers who demand high wages will be required to make the repairs.

Wastage and Stock Reserves

If we stopped to realize it, under the present system, a large number of units and parts are actually wasted because a great deal of material can be reclaimed at a very substantial profit if properly handled and managed. Standard crates and packing boxes could be designed for making shipments convenient and efficient. In many cases the same crates and boxes or containers can be made for re-shipment of definite material in the event the equipment is standardized. This will simplify shipments and encourage repairs by the unit-replacement system. The greatest handicap to the unit-replacement system is the inconvenience caused and the management required for procuring replacements and disposing of the unserviceable units.

No matter where the units are rebuilt, the factory service-stations, feeders and distributors should carry a reserve stock of units for replacement. If unit assemblies are rebuilt in accordance with the economical and efficient production-methods and the laws of management already listed, the factory charge for exchanging an unserviceable unit for a serviceable unit will be so small that the fleet operator or service-station manager will discontinue the repair of units in their own organizations.

It is very doubtful if units can be rebuilt economically at factory service-stations unless each station has control and jurisdiction over a very large territory so that sufficient unserviceable units will be received.

I have visited several factory service-stations where units are rebuilt and have often observed special machinery and equipment used for repairing units that are idle and occupy valuable floor-space. The floor space that these machines require is an expensive item. There are very few fleet operators who can rebuild unserviceable units in mass quantities. Unless machine-tools and equipment are used constantly, the overhead expense will increase the cost of repairs unnecessarily. Further, the plant in which the units are rebuilt must be organized into departments so that the work is divided.

The following fundamental principle of management must be kept constantly in mind: One or only a few mental operations should be assigned to each worker. This tends to improve and greatly increase the quality of the output. The engine division, for example, should be subdivided into the following departments: (*a*) cylinder grinding, (*b*) crankshaft grinding, (*c*) crankcase-bearing section, (*d*) connecting-rod-bearing section, (*e*) piston grinding, (*f*) piston-pin grinding, (*g*) valve grinding and (*h*) camshaft grinding. High wages do not need to be paid when a worker is required to perform only one operation. This also simplifies the training of men.

Accessory Units Considered

The accessory units such as the fan, carbureter, ignition unit, vacuum pump, water pump, oil pump, ignition switch, ammeter and the like, are all rebuilt in the accessory-units department which, in turn, is subdivided so that maximum efficiency and minimum cost can be obtained. If units are rebuilt in this manner and the worker and his equipment are kept busy constantly, the cost of repairs will be reduced greatly. After an unskilled mechanic has become familiar with the mental or manual operations required, he produces the same quality and quantity of output as a skilled mechanic. This will make it possible to get satisfactory skilled work done without employing skilled mechanics who can demand a high wage.

It is very doubtful whether manufacturers of motor-vehicles desire to adopt the unit-replacement system as this system for making repairs will, no doubt, increase the life of a motor-vehicle and therefore reduce sales. The principal factor which will determine when a vehicle should be discontinued from service is obsolescence.

Gasoline Requirements of Commercial Aircraft Engines[1]

By H. K. CUMMINGS[2]

ST. LOUIS AERONAUTIC MEETING PAPER *Illustrated with* CHARTS

THIS paper was prompted by the numerous inquiries received by the Bureau of Standards from airplane owners and airport operators regarding the grades of gasoline that are suitable for aircraft use and the suggestion that the Bureau test all brands of commercial aviation gasoline and publish the results or that the Department of Commerce issue approved type certificates for certain standard grades of aviation gasoline.

The purpose of the author in presenting the paper was to open discussion on the subject, and in this he was very successful.

The problem of a suitable fuel and its general distribution throughout the Country for aviation use is complicated by considerable divergence of opinion among aircraft-engine manufacturers as to the kind of fuel preferred and by wide differences in detonation characteristics of gasolines of like volatility. No chemical or physical test has been found that will serve as a sure criterion of antiknock value, and laboratory tests on the same group of fuels in a variety of engines have given results that are not entirely comparable.

The subject is summed up in the concluding statement that aviation gasoline should have adequate antiknock value and volatility and should not contain an excess of gum or sulphur. It seems to be the engine manufacturer's problem to make knock tests and determine how volatile a fuel is required for his engine. The engine manufacturer's problem would be greatly simplified if more oil companies would adopt the policy that has been adopted by at least two oil companies of marketing all over the Country, under definite trade names, aviation gasolines of essentially constant quality as regards antiknock value and volatility.

Discussers representing oil companies and aircraft and engine manufacturers present their respective difficulties with the problem, and the long-standing dispute over the responsibility of finding a solution is renewed. The oil men assert that it is the task of the engine builders to design engines that will operate satisfactorily on the available fuels, and the engine makers claim that the oil companies should produce and distribute universally a fuel that will not knock in high-compression engines.

INQUIRIES received by the Bureau of Standards from airport operators and airplane owners regarding the grades of gasoline which are suitable for aircraft use indicate considerable uncertainty as to the gasoline requirements of present-day commercial aircraft engines. The suggestion has, in fact, been made that the Department of Commerce test all brands of commercial aviation gasoline and publish the results from time to time, or else that "approved type certificates" for certain standard grades of aviation gasoline be issued. While no such regulation of the oil industry seems to the Department to be necessary or desirable, it is proper to examine the situation and see what assistance can legitimately be given the purchaser of aviation gasoline, since an unwise choice of fuel may indeed cause a forced landing or even a fatal crash.

A questionnaire recently sent to the manufacturers of approved engines showed that about one-half of the engine builders agree with the distinguished editor of *Aviation* in regarding domestic aviation gasoline as a "standard fuel on sale at every airport" and recommend without material qualification the use of this fuel in their engines. The remaining manufacturers endorse particular brands of domestic aviation gasoline or recommend gasoline treated with lead or benzol where California domestic aviation gasoline is not available. When asked if motor fuel could be used with safety in case it was necessary to refuel where no aviation gasoline could be obtained, several manufacturers of relatively low-compression engines stated that their engines performed well on ordinary motor gasoline. However, most of the manufacturers recommended for emergency use only premium automobile fuels, preferably those which are both high test and high compression, and advised part-throttle operation with mixture control full rich. If wide-open throttle is required on the take-off with a low-grade fuel, detonation may be minimized by retarding the spark.

It is well known that gasolines of like volatility may differ widely in their detonation characteristics, straight-run gasolines from certain eastern petroleums being among the worst "knockers," whereas California gasolines are generally superior in antiknock value. Unfortunately, no chemical or physical test has been found which will serve as a reliable criterion of antiknock value, and actual engine tests are required to determine whether a particular gasoline will knock in a given engine. Different laboratories use a variety of test engines and test procedure in estimating the relative tendency of different gasolines to knock. Tests by a number of laboratories on the same group of fuels have shown that the results thus obtained are not entirely comparable, and efforts are now being directed toward the development of a standard test-engine and procedure. The general adoption of a uniform knock-test will make it possible to specify definitely the detonation characteristics of fuels which are found to give satisfactory performance in any engine.

[1] Publication approved by the Director of the Bureau of Standards of the United States Department of Commerce, City of Washington.

[2] S.M.S.A.E.—Physicist, chief, automotive powerplants section, Bureau of Standards, City of Washington.

Knock Tendency a Function of Engine Design

Recognizing that the tendency of aviation gasolines to knock in a given engine is a function of engine design—compression ratio, combustion-chamber shape, spark-plug location, cylinder-head temperature, distribution and the like—the Department of Commerce expects the engine manufacturer to determine by trial what grades of gasoline are satisfactory in his engine. When a new type of engine is submitted for official test, the manufacturer is required to supply a suitable fuel, and the engine, if successful, is approved only for use with fuels equal or superior in antiknock value to that used during the official test. For this purpose the antiknock value is expressed in terms of the approximate percentage of benzol which must be blended with an eastern domestic aviation gasoline (the Bureau of Standards' current supply) to match the test fuel in a high-compression single-cylinder Liberty engine. The apparatus and test method have been described previously[3]. Nearly all the test fuels supplied are distinctly superior to this reference fuel.

The Federal specifications for aviation gasoline and the methods of test employed are described in Bureau of Mines Technical Paper 323 B. They contain no requirement as to antiknock value, for reasons already explained, but the military services, which are the chief purchasers under these specifications, are authorized to impose supplementary tests pending the establishment of a standard knock-test. The specifications are designed to accomplish two purposes: (*a*) to secure a product of suitable volatility and (*b*) to guard against harmful amounts of gum and sulphur.

Motor Fuel and Aviation Gasoline Compared

The distillation ranges of the two motor-fuel specifications and the two aviation-gasoline specifications are shown for comparison in Fig. 1. Curve *A* represents the new U. S. motor gasoline in which a 10-per cent-point specification has taken the place of the initial and 20-per cent points formerly specified. The maximum 10-per cent point is normally 80 deg. cent, (176 deg. fahr.), but in cold weather this may be reduced to 70 deg. cent. (158 deg. fahr.) if the purchasing officer so elects. Curve *B* represents high-test motor fuel for special service, as in ambulances and fire apparatus. Prior to the drafting of this specification, the District of Columbia Fire Department used domestic aviation gasoline. The maximum 10-per cent point of this fuel is 65 deg. cent. (149 deg. fahr.), whereas both the fuels represented by Curves *A* and *B* have minimum 10-per cent points at or below 50 deg. cent. (122 deg. fahr.).

The aviation-fuel curves *C* and *D* have 50-per cent points and 90-per cent points distinctly below the motor-fuel curves, but compare unfavorably at the lower ends, for maximum 5-per cent points at 75 deg. cent. (167 deg. fahr.) and 65 deg. cent. (149 deg. fahr.) correspond approximately to 80 deg. cent. (176 deg. fahr.) and 70 deg. cent. (158 deg. fahr.) at 10 per cent. The minimum 5-per cent points at 50 deg. cent (122 deg. fahr.) correspond to about 60 deg. cent. (140 deg. fahr.) at 10 per cent. Since the 10-per cent point is the criterion of starting ability[4] as well as vapor lock, these points will doubtless be lowered whenever the aviation-gasoline specifications are next revised.

[3] See S.A.E. JOURNAL, May, 1927, p. 605, and April, 1928, p. 455.

[4] See S.A.E. JOURNAL, March, 1927, p. 353, and October, 1929, p. 345.

[5] See S.A.E. JOURNAL, November, 1928, p. 478.

The other specification point of major importance is the 90-per cent point, which has been shown[5] to determine the dew-point, or the minimum temperature for complete vaporization, of the gasoline in the presence of air. The maximum 90-per cent temperatures of 155 deg. cent. (311 deg. fahr.) and 125 deg. cent. (257 deg. fahr.) correspond to dew-points of 33 deg. cent. (91 deg. fahr.) and 11 deg. cent. (52 deg. fahr.) for air-gasoline mixtures containing 12 parts of air to 1 of gasoline by weight. As indicated in the references cited, it is preferable to allow for distillation loss by plotting the distillation curve in terms of "per cent evaporated" instead of "per cent distilled" as is now customary.

Curve *A* in Fig. 2 shows an average distillation curve for motor gasoline based on 162 commercial samples collected throughout the Country in July, 1929, for the 20th Bureau of Mines semi-annual gasoline survey (R. I. Serial 2959). Distillation curves for three commercial aviation gasolines are also shown, together with the aviation-gasoline specification points. Curve *B* represents a domestic aviation gasoline which is probably below the average in volatility. Curve *C* represents an aviation gasoline of fighting grade having excellent starting volatility but the maximum dew-point for this grade. Curve *D* represents one of the stabilized natural-gasoline products which have been used by several of the air-transport companies during the last year or two. A study of data on British and European aviation gasolines furnished by the Standard Oil Co. of New Jersey and the Vacuum Oil Co. indicates that they average close to the upper limit of our fighting-grade specification and are inferior in starting volatility to the gasoline represented by Curve *C*. The average curve is 9 per cent off at 65 deg. cent. (149 deg. fahr.),

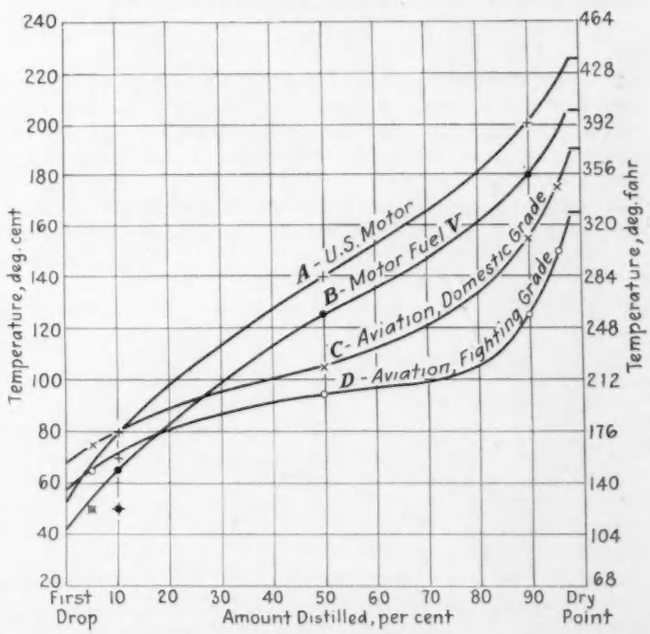


FIG. 1—DISTILLATION RANGES OF FEDERAL SPECIFICATIONS FOR TWO MOTOR FUELS AND TWO AVIATION GASOLINES

Curve *A* Represents the New U. S. Motor Gasoline in Which a 10-Per Cent-Point Specification Has Taken the Place of the Initial and 20-Per Cent Points Formerly Specified. Curve *B* Represents High-Test Motor Fuel for Special Service. Curves *C* and *D* for Aviation Gasolines Have 50-Per Cent and 90-Per Cent Points Distinctly Below the Motor-Fuel Curves but Compare Unfavorably at the Lower Ends

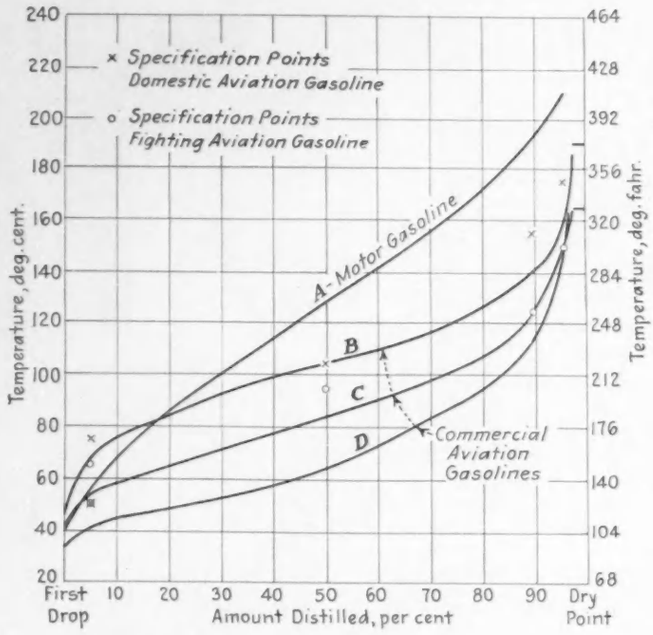


FIG. 2—A.S.T.M. DISTILLATION CURVES FOR MOTOR GASOLINE AND COMMERCIAL AVIATION GASOLINES

Curve *A* Is an Average Based on 162 Samples Collected throughout the Country in July, 1929. Curve *B* Represents a Domestic Aviation Gasoline That Is Probably Below the Average Volatility. Curve *C* Represents Fighting-Grade Aviation Gasoline Having Excellent Starting Volatility. Curve *D* Represents a Stabilized Natural Gasoline

54 per cent off at 100 deg. cent. (212 deg. fahr.) and 90 per cent off at 125 deg. cent. (257 deg. fahr.).

Gum and Sulphur Tests Inconclusive

The collection of gum on inlet valves and valve-stems where gasoline spray mixed with air impinges on hot metal is likely to occur if fuel containing excessive amounts of gum is used in the engine. The specification requirement that the residue, when 100 cc. of gasoline is evaporated in a polished copper dish, shall not exceed 3 mg. is probably effective in keeping out such gasoline. This test, however, is lacking in reproducibility and doubtless eliminates many gasolines which do not contain harmful amounts of gum. The gum problem has grown in importance with the increased production of cracked gasoline, and the present test is likely to be modified or replaced as the result of studies now in progress at the Bureau of Standards and in several other laboratories.

Gray or black discoloration of the polished copper dish mentioned is taken to indicate the presence of corrosive sulphur compounds. Even where this test is negative, gasolines containing more than 0.10 per cent of sulphur are ruled out, since the products of combustion include sulphur dioxide and water vapor, which combine at low temperature to form sulphurous or sulphuric acid in the crankcase. Refiners working with petroleum especially high in sulphur argue that limitation of the sulphur to 0.10 per cent is unduly severe, but the Bureau of Standards plans a comprehensive study of this matter of sulphur corrosion as soon as Government funds are available for the purpose.

To sum up the question of aviation-gasoline specifications, the fuel should have adequate antiknock value and volatility and should not contain excessive gum or sulphur. Pending agreement on a standard test, antiknock value can be estimated by actually operating the engine on different fuels; volatility data can be deduced from the A. S. T. M. 10-per cent and 90-per cent points. The present test for gum is unsatisfactory but will serve, pending further investigation of the problem, provided the gasoline is used soon after the test is made; and the present sulphur limitation is admittedly on the safe side. It appears to be the manufacturer's problem to make knock tests and to determine how volatile a fuel is required for his engine. Any commercial testing laboratory can make the laboratory tests described in Bureau of Mines Technical Paper 323B.

If more companies were to adopt the policy already adopted by at least two oil companies, of marketing all over the Country, under definite trade names, aviation gasolines of essentially constant quality as regards antiknock value and volatility, the problem of the engine manufacturer in determining what aviation gasolines he should recommend to his customers in different sections of the United States would be greatly simplified.

THE DISCUSSION

CHAIRMAN T. A. PECK[6]:—I think that Mr. Cummings has summarized the problem that is of vital importance to the engine manufacturer, the airplane manufacturer, the transport company and the refining industry. His concluding statement at least will stand out in my mind; if more companies would follow the example of those two pioneers who have already given the aircraft industry of America a uniform and dependable fuel throughout the Nation, the work of the aircraft-engine designer, the transport manufacturer and the transport-company officials would be much simplified. It certainly is a great disappointment to start off complacently from Portland, Me., as I did last year, with an excellent fuel on which the engine functioned beautifully and at the Jersey City airport to be introduced to something I did not even recognize; then to get out into the Middle West and find still another fuel, and so on until I got across the ridge. It is really a problem worth considering.

Mr. Cummings has said that no chemical or physical test is a criterion of the highest useful compression-ratio of fuel, which we have to admit is something we are all working for and worrying about as we are trying to get more power from these engines. He also said a little later that the H. U. C. R. is a function of engine design. We in the oil industry know that it is true. The fuel with an H. U. C. R. of 5.8 in one engine will have 5.2 or 5.3 or 5.4 in another.

T. L. ROBINSON[7]:—I should like to ask Mr. Cummings what the probability is of a basic corrosive agent being present rather than the sulphuric acid. Is there anything that might possibly produce this reaction other than the products of combustion?

H. K. CUMMINGS:—I do not think it is possible to

[6] M.S.A.E.—Consulting engineer, St. Louis.
[7] M.S.A.E.—Chief engineer, McGill Metal Co., Valparaiso, Ind.

give a definite negative answer to that, but it is quite obvious that, if there is sulphur in the fuel, the products of combustion will include at least sulphurous and probably sulphuric acid. The study of corrosion is a sufficiently complicated problem and so little work has been done on it that it is impossible to say what other corrosive actions may occur; in the automobile engine it is the corrosion of iron; in the aircraft engine it is generally the corrosion of aluminum alloys, which is more serious in alkaline solutions.

MR. ROBINSON:—My question was prompted by the fact that in some cases aluminum-bronze valve-inserts have been very rapidly attacked and it is difficult to see how that corrosion could occur under acid conditions, because aluminum bronze is a very good anti-corrosive alloy under acid conditions.

Fuels Selected by Author for Comparison

ATHEL F. DENHAM[8]:—Were the samples of the fuels *B*, *C*, and *D* that Mr. Cummings showed in Table 2 taken at random, or were they picked for poor grade, medium grade and high grade? How they were selected might have some bearing on the subject.

MR. CUMMINGS:—Those fuels were selected from distillation data that we had for other purposes. One was a commercial fuel from one of the airports. One was an experimental fuel sent in but which agreed very closely with published data. The *D* fuel was a natural gasoline; that happened to be Naturaline, and it is not very far out from the published figures on the fuel that Boeing has been using, a Phillips Petroleum natural gasoline. I cannot give offhand the sources of *B* and *C*. Of the three aviation gasolines, one was taken as representing more or less the upper limit of volatility of domestic aviation gasoline; the next one represented somewhat less than the upper limit of the fighting grade, and the third exceeded in volatility the fighting grade.

The impression I have is that most engines will gain comparatively little improvement from volatility increased beyond that of fighting grade. There may be some improvement, but the question of high-volatile gasoline is largely an economic one. If the aviation natural gasolines compete in price with the fighting grade obtained by ordinary distillation processes, they may well have a large use, but there will evidently be a point in the design of a particular engine beyond which increase in volatility will not warrant an increase in the cost of the fuel. That was the point I had in mind.

G. G. OBERFELL[9]:—I noticed that sample of Naturaline on the chart. I would not agree with Mr. Cummings that there is very little difference between it and the Phillips Petroleum Co.'s products, primarily for the reason that our aviation natural gasoline has an initial boiling-point of 100 to 101 or 102 deg. fahr., while it looks as if the initial boiling-point of the sample was probably 10 deg. lower than that. Also, the 5-per cent point looks to me as if it were about 104 deg., and our 5-per cent point is 112 to 114 deg. We found in actual flying operations that there is a decided difference in natural gasolines for aviation fuel between those two grades from the standpoint of actual test. We have manufacturing specifications that preclude anything below 100-deg. initial and a 5-per cent point of 112 to 114 deg. We have had an experience now of two or three years with a large number of airplanes that have flown millions of miles. While I dislike to take exception to Mr. Cummings' statement, I think our aviation gasoline, from the standpoint of actual use, is somewhat different.

Dr. Bridgeman has shown that a small percentage of propane is responsible to a very great extent for the increase in vapor pressure of motor fuel. As a matter of fact, our percentage of propane will contribute about 2 lb., whereas 1 per cent of butane will contribute probably about 0.5 lb. The ratio is about 4 to 1, but in ordinary aviation gasolines one will find only about ½ per cent of propane and about 4 per cent of butane. So, while the propane for an equal percentage may be about four times as bad, or vicious, the actual presence of butane is such that it contributes about eight times as much as does propane to the vapor pressure of ordinary aviation gasoline. Our aviation natural gasoline is practically devoid of butane. For that reason we can put a larger percentage of pentane into it and have a high volatility at a lower vapor-pressure.

Relative Effect of Tetraethyl Lead

ANSON M. KELLER[10]:—Has Mr. Cummings any information available on the relative effects of different concentrations of tetraethyl lead on the knock properties of gasoline for use in aviation engines?

MR. CUMMINGS:—Only general statements can be made. The effect of adding tetraethyl lead to a given gasoline appears to decrease noticeably after some limited amount has been added. However, the amount of improvement from a given addition of lead to various gasolines may be quite different. Each gasoline is more or less a special case, and lead is not particularly desirable as a standard criterion of how good a gasoline is. In general, so far as we can judge, a more or less uniform benefit is obtained from small additions of lead, but if the process is continued a point is reached at which the same increment of lead produces very little benefit. That is likely to be outside the range of anything that one would want to use in practice.

MR. KELLER:—Approximately what would that concentration amount to in the average case?

MR. CUMMINGS:—Ordinarily, not more than 3 milliliters of tetraethyl lead or 5 of ethyl fluid are added per gallon, and in adding that amount to some gasolines one might reach the point of limiting return, whereas with others he might still be getting equal effect from successive additions. Hence no definite answer to your question is possible. Some observers have reported that California gasolines receive more benefit from given additions of lead than some of the eastern gasolines, but that may not be generally true.

Permissible Gum-Content Uncertain

T. B. RENDEL[11]:—You raised the question of the permissible gum-content in gasolines. Are you prepared to amplify that statement a little? The specifications call for a maximum of about 3 mg. per 100 cc. of gasoline, but it is my opinion that we could go a little higher than that, and I should like to know what your maximum point on this is?

MR. CUMMINGS:—I have no definite figures in mind. I agree with anybody that the test is merely an expedient. I think it is safe to say that the copper-dish test probably rules out many gasolines that are per-

[8] Field editor, *Automotive Industries*, Detroit.

[9] Director of research, Phillips Petroleum Co., Bartlesville, Okla.

[10] Engineer, Ryan Aircraft Corp., St. Louis.

[11] Jun. S.A.E.—Motor testing, research engineer, Shell Petroleum Corp., Wood River, Ill.

fectly good. It is successful in eliminating the gasolines which contain harmful amounts of gum. So far as my information goes, the residue in the copper-dish represents the gum already present plus an unknown contribution from ingredients that may produce gum under storage conditions, but it does not give either an accurate figure of the preformed gum or of the potential gum. The present test has a factor of safety if the gasoline is not placed in storage after test, but I would not venture any statement as to how much it would be safe to increase the 3-mg. limit.

CHAIRMAN PECK:—In that connection would you care to express your opinion on the gum test as to the color of the deposition? I lean to the opinion that if the deposition is aluminum colored or silver colored it is possible to have a much larger content without any apprehension than it is if the deposition is black.

Mr. CUMMINGS:—I have nothing definite to contribute to that. I suppose the idea is that, in the absence of the dark color, the absence of corrosive sulphur is indicated. I think it is quite possible that it would be safe to increase the 3-mg. limit under such circumstances, but it could hardly be done without an experimental basis. Any modification or the replacement of the present test will probably await the completion of experimental work that the Bureau of Standards is doing for the Army, and will be based on all available experimental results.

CHAIRMAN PECK:—Does the Bureau contemplate any work along those lines?

MR. CUMMINGS:—The Bureau is making a study of the gumming characteristics of aviation gasolines at the request of the Army and that work has recently been taken on by Dr. Bridgeman.

Bureau Working on Accelerated Gum-Test

CHAIRMAN PECK:—Does anyone here know anything about the acceleration test that is being carried on at Wright Field? Has anyone had any experience with it or adopted the methods of the Government at Wright Field in finding out what is going to happen in gasoline after it has been in storage six months? I understand that very responsible companies that are vendors to the Government have at times delivered aviation gasoline that is perfect in quality when delivered and in a few months becomes of different quality. Has the Bureau done anything or had any experience with the outfit that is set up at Wright Field?

Mr. CUMMINGS:—The Bureau is working on accelerated tests for potential gum as well as on comparative tests of gum by the copper-dish, steam-oven and the other methods that have been started in other laboratories. My impression is that the test you have reference to is very similar to the one the Standard Oil Co. of New Jersey uses[12].

CHAIRMAN PECK:—Is the Bureau now working on that, and what pressure is applied, if it is a fair question?

MR. CUMMINGS:—The initial oxygen pressure was 6 or 8 atmospheres, and the bomb was placed in a steam bath. Various pressures and temperatures will be used in future tests.

W. E. SCHWARZMAN[13]:—Mr. Cummings, in the beginning of your talk you spoke about retarding the spark to avoid or to help along certain conditions. How about those instances when fixed ignition is resorted to, where the spark always occurs at full advance regardless of speed or load?

Compression Kept Low Because of Fuels

MR. CUMMINGS:—In such cases you cannot use that particular expedient. That was simply quoted as the suggestion of one manufacturer for meeting an emergency when motor fuel has to be used and it is not possible to take off with the engine throttled. In the event that the spark cannot be retarded, the load may be reduced so as to take off, or it may be possible, as another manufacturer suggested, to buy some benzol at a drug store and put that into the gasoline, thereby improving its antiknock qualities enough to make possible the take-off.

It is not true that the majority of commercial aircraft engines are designed to operate at full load on any grade of motor gasoline; it is true, however, in view of the uncertainty of getting high-antiknock aviation gasoline at all points, that many designers use lower compression-ratios than they might otherwise be disposed to use. I think the situation is a good deal similar to the motor-fuel situation of a few years ago; that is, the automobile manufacturers for many years kept their compression ratios below 5:1. After it became evident that quite general distribution had been made of gasolines permitting higher compression-ratios, certain manufacturers ventured to increase the compression ratios, and one manufacturer went farther than was found entirely satisfactory and dropped back a little. But the result has not been that all the automobile-engine compression-ratios are now between 5 and 6; they are still between 4 and a little above 5. The majority of engine manufacturers are satisfied to offer as alternative choices special heads or special compression-ratios, and it is doubtful whether, even if it were known that aviation gasoline permitting the use of a compression ratio of 6.5:1, we will say, was generally available, all the engine manufacturers would hasten to take advantage of that fact.

The use of increased compression-ratios promotes fuel economy but entails the use of suitable fuels; that is, damage may result when suitable fuels are not available, so the high-compression engine requires a supply of fuel of special antiknock value. The present trend in the Military Service is to increase the compression ratio on experimental engines flying in experimental planes. The Service in general probably will follow less rapidly, and my own guess is that, even if fuels of relatively high antiknock value are available, it will be found that many aircraft engines for many years to come will have comparatively low compression-ratios. That is along the line of conservatism and along the line of motor-car experience.

Better Results from Slower-Burning Fuels

E. B. HEATH[14]:—Your remarks about compression ratios brings up a very interesting subject. I think the oil companies will find that it is they who are holding down the compression ratios. We make a few airplane engines. Every automobile manufacturer is rather well satisfied that if he can get the right fuel he will increase the compression ratio as much as it will permit. About half the automobiles running on the streets today will not run well with the cheaper grades of fuel we buy. From my own experience in driving a

[12] See S.A.E. JOURNAL, January, 1930, p. 31.

[13] M.S.A.E.—Engineering representative, American Bosch Magneto Corp., Springfield, Mass.

[14] President, Heath Aircraft Corp., Chicago.

car, I find it is much cheaper to use a slow-detonating fuel, either ethyl or California gasoline, preferably ethyl if I can get it, because I get about 5 m.p.h. more out of the car, and it is the extra five that we need; and in getting that extra 5 m.p.h. we also get considerable improvement in miles per gallon.

I think that is a very important point to consider. It is the oil men, not the airplane men, who are holding back all the gasoline-engine performance today. If we could buy the slower-burning fuels and if the general public knew that they really give the proper results, I believe more of them would be used.

If the regular aviation gaoline or any of the rapidly burning fuels is put in the new Model-A Ford car, the engine will knock when the car is stopped and then suddenly accelerated. The compression ratio is right up to the ragged edge which gives the highest possible performance. Mr. Ford would put it a little higher, and I believe we would all put it higher, if we could get the fuel to do it with. I think that all the fuel that is sold should be slow-burning, and it is the responsibility of the oil companies to establish just how far we shall go in the line of a slow-burning fuel.

Uniformity of Characteristics Most Important

MR. CUMMINGS:—I believe it is a question of how willing the engine manufacturers are to experiment; whether they want to take it all at a jump or, as the fuels are improved, the general trend will be rapid or gradual. The more significant matter, it seems to me, is that to which the Chairman called attention; namely, the fact that in some cases a concerted effort is being made to assure that a particular brand of gasoline has, as far as the oil company can control it, essentially the same volatility characteristics and the same detonation characteristics wherever it is sold. That is something to tie to and that seems to me of more importance in the aircraft industry than in the motor-car industry even, because the distances covered by the airplane are greater and the pilot is likely to be seeking fuel in different parts of the Country.

That is no small thing for the oil companies to do. I assume that many engine men fail to realize the fact that the high quality of California gasoline is a high average and individual gasolines from California crudes may differ nearly as much in antiknock value as individual gasolines from Mid-Continent crudes, and that only by selection or some other control method could a uniform antiknock value be maintained for a particular brand of aviation gasoline. It is a large order for the oil men to maintain and distribute widely fuel which is uniform not only in distillation characteristics but in detonation characteristics.

High-Compression Engines Very Fuel-Sensitive

BURNHAM ADAMS[15]:—The company with which I am associated has on several occasions put out engines of very high performance for some special occasion. We have been very careful in every case to make sure that the operator understood clearly that the engine was not a service engine and that he had something of which he had to be very careful. I know of at least two instances in which, in spite of all precautions, there have been very serious engine failures because the pilot got caught short of fuel where he had to use an ordinary aviation gasoline in a 6.5:1-compression-ratio engine.

The problem boils down to a matter of the commercial distribution of fuel. High-antiknock gasoline can be used with entire satisfaction in the 5.0 or 5.2:1 engine, but aviation gasoline of the present standard cannot possibly be used in 6.0 or 6.5:1 engines without blending. This indicates the need of first making the fuel available, then following it up with the engines.

We have found in recent tests that we can put 8:1 pistons and 8:1 supercharger gears in a 525-hp. engine and get 620 hp. out of it at rated r.p.m. Then, with a 5-per cent drop in power, the fuel consumption can be reduced to 0.41 lb. per hp. In other words, we can get a 20-per cent increase in power and a 20-per cent decrease in fuel consumption by going to higher compression and higher supercharging, but we cannot do it on anything less than the present aviation gasoline blended with about 60 per cent of benzol. The Army and the Navy are very much interested in this point. Both are planning on going to higher compression on Service engines within the next year. They have new specifications out and I believe are already buying gasoline on a 5.8 H.U.C.R. rating. These engines will be kept in rather restricted areas where they can be watched.

The sandpaper squad has the modern engine sandpapered down to about the minimum weight. It is going to be a long, hard pull to further reduce the weight per inch of displacement, so the increase in power during the next two years will have to come through increased crankshaft speeds and increased compression and supercharging. To do that we must have higher-compression fuels. This is not so very far away, as far as engine manufacturers are concerned. If the fuel is available where the commercial operator can get it, the engine manufacturers can have the engines available for service on very short notice.

Fuel Systems Unsuited to High Volatility

Dr. RAYMOND HASKELL[16]:—There are three or four points I want to bring up. I think Mr. Cummings has been too modest regarding the two different projects with which the Bureau is connected. One is the standardization of an antiknock engine, which we hope in the next six months will be complete; for this purpose the Bureau is now, I think, just going through and checking up about ten different laboratories that are trying out this engine so that we can get a standard engine that will give us better results. I think Mr. Cummings will agree that when we get around 50 or 60 benzol equivalent the engine is touchy. There are bigger unit differences in antiknock value around 50 and 60 benzol equivalent than there are around 15 and 20; and when we get up to the high figures we do not know exactly where we stand; the different engines will act differently.

With regard to gumming, a committee has been formed in which the Bureau of Standards has an important part and which includes representatives of the aviation and the oil industries and all others interested in the study of gumming. This is one of the principal properties of gasoline that they want to study now, particularly as we are going to the high-antiknock gasolines. Until we get our refining methods a little more straightened out there is a chance for gum. We do not know just yet what causes it, when it is harmful and when it is not. I know the Army has run as high as 10 and 15 mg. of gum with entire satisfaction, while in

(*Concluded on p.* 233)

[15] Jun. S.A.E.—Engineer, Wright Aeronautical Corp., Paterson, N. J.

[16] Technical division, The Texas Co., New York City.

The Vapor-Locking Tendency of Aviation Gasolines[1]

By Oscar C. Bridgeman[2] and Hobart S. White[3]

St. Louis Aeronautic Meeting Paper — *Illustrated with* Charts

PREVIOUS reports on the investigation of vapor lock in airplane fuel-systems have indicated that the tendency of aviation gasolines to cause trouble from vapor lock could be predicted from the standard A.S.T.M. distillation-data on the gasolines. The present paper deals with experiments made for the purpose of comparing the predicted conditions for vapor lock with those actually found in typical fuel-feed systems to cause trouble from this source. In gravity-feed systems, the predicted and experimentally observed conditions for vapor lock are in good agreement. On the suction side of a fuel pump, vapor lock occurs in a number of cases at lower temperatures than those predicted, particularly under conditions representing high-altitude flying in a system employing a long suction-lift.

THE fuel-feed system in any automobile or airplane serves to transport the fuel from the supply tank to the intake manifold of the engine. Specific designs of fuel systems may vary but, in every case, their common function is to deliver the fuel continuously to the engine in the desired amounts if uninterrupted operation is to be obtained. A well-designed fuel-system is one which will perform this function satisfactorily under all conditions of operation with a wide variety of fuels. A system which is "well designed" for use on one particular grade of fuel might be classed as "poorly designed" when used with a more volatile fuel. This means that a fuel-feed system should not be designed without regard for the type of fuel which will be used.

The more volatile a fuel, the greater will be the tendency to boil in the fuel-feed system. If boiling occurs, then interruption of flow due to vapor lock may be expected. The property of a gasoline which determines the tendency to boil is the temperature at which 10 per cent of the gasoline is evaporated in the standard distillation-test. For an automobile fuel-system, the temperature of the fuel should not exceed the 10-per cent point, if vapor lock is to be avoided. For airplane fuel-systems, the gasoline temperatures should be kept below this point by about 2 deg. fahr. for every 1000 ft. above sea level. Hence a gasoline with a 10-per cent point of 140 deg. fahr. should not exceed 100 deg. fahr. in an airplane at 20,000 ft. Fuels containing undesirable amounts of propane may cause trouble at temperatures considerably lower than those predicted from the 10-per cent points.

Fuel systems employing long suction-lifts may be subject to vapor lock at lower temperatures than those employing a straight gravity-feed. On the suction side of the pump, the gasoline is under reduced pressure which increases the tendency to vapor lock. Accordingly the suction lift with a fuel pump should be as short as possible. Apparently minor details of design and installation may have a very marked effect on vapor lock. Changes in T-connections, increase in size of tubing, elimination of unnecessary bends and fittings and rearrangement of the bypass line around the fuel pump may add many thousands of feet to the ceiling of an airplane and may mean the difference between a completed trip or a forced landing.

Some of the precautions regarding fuel lines and fuel which should be observed in order to have satisfactory airplane operation are as follows:

(1) The fuel lines should be so designed and installed that the temperature of the liquid fuel does not exceed the point at which the vapor pressure of the gas-free gasoline equals the external pressure. This can be computed from the 10-per cent distillation point. In meeting this requirement, a reasonable margin of safety should be allowed for variations in existing fuels.

(2) Long suction-lifts by a fuel pump should be avoided. If the height between the tank and the carbureter is more than 2 to 3 ft., greater safety of operation and increased efficiency can be obtained by using a flexible drive.

(3) Use of a fuel more volatile than necessary for easy starting increases the tendency to vapor lock. Fuels containing more than small amounts of propane are undesirable.

In a paper[4] presented at the last Annual Meeting of the Society, data were given on the properties of gasolines which affect their vapor-locking tendency. This work led to the conclusion that the vapor pressure of the gas-free gasoline was essentially the only property of a gasoline which is of interest as regards vapor lock in a well-designed fuel-feed system. From this study, it was possible to predict the conditions under which any given fuel would cause trouble from vapor lock. The present paper covers a second phase of the investigation in which the predicted conditions for vapor lock are compared with those actually found to interrupt fuel flow in two types of airplane fuel-feed systems. For the gravity-feed systems studied, there is remarkable agreement between the predicted limiting conditions and those actually found in the flow experiments to cause vapor lock. In the fuel-pump experiments, the

[1] Publication approved by the Director of the Bureau of Standards of the Department of Commerce.

[2] S.M.S.A.E.—Research associate, Bureau of Standards, City of Washington.

[3] Junior physicist, Bureau of Standards, City of Washington.

[4] See S.A.E. Journal, July, 1930, p. 93.

observed vapor-locking temperatures were found to be lower than those predicted, by amounts dependent upon the height of suction lift, the speed of the pump and the altitude.

Outline of Vapor-Lock Investigation

Vapor lock is the partial or complete interruption of gasoline flow in the fuel-feed system, resulting from the formation of bubbles of vapor or gas. It is frequently called "gas lock" or "air lock." When vapor lock occurs, the behavior of the engine is analogous to that observed when water or dirt in the fuel line impedes the flow.

Vapor lock may result due to the use of too volatile a fuel, or to overheating of the fuel in the feed system, or to employment of excessive suction-lifts with a fuel pump. The more volatile a fuel is, the more readily will vapor lock occur in any given system. On the other hand, the tendency to vapor lock is increased by raising the fuel-line temperature or by flying at higher altitudes. Further, long suction-lifts by a fuel pump are more frequent causes of trouble from this source than are short lifts.

A fuel can be produced which will cause vapor lock in any fuel system. Conversely, a fuel system can be designed which will result in vapor lock with any fuel. The problem of vapor lock involves both the aircraft and petroleum industries, for both can do much to minimize this source of hazard in flying. The solution to the problem lies in the cooperation of both industries in producing the most satisfactory type of fuel for use in an engine equipped with the optimum design of fuel-feed system.

Since fuels and fuel-line design are both involved in a complete investigation of vapor lock, the problem was divided into four phases:

(1) Study of the properties of gasolines which determine their vapor-locking tendency

(2) Determination of the actual conditions under which vapor lock occurs in typical fuel-feed systems in the laboratory

(3) Evaluation of the effect of design factors on ease of vapor lock

(4) Measurement of the temperatures existing in the fuel-feed systems of representative aircraft during flight

The first phase of the investigation, reported at the last Annual Meeting of the Society, has given a means for predicting the conditions under which vapor lock may occur and has furnished the basic information for the interpretation of the data obtained in the second phase of the work, which constitutes the present paper. The third phase, on design factors, is just being started and much suggestive material has been obtained to date regarding certain phases of this study which merit investigation. Some data have also been obtained on the temperatures in airplane fuel-feed systems under flight conditions, but a report on this is being withheld until a more representative group of data have been secured.

Predicted Conditions for Vapor Lock

From the standpoint of the fuel, serious interruption of flow due to vapor lock under any given set of conditions is dependent upon the vapor pressure of the gas-free gasoline and upon the volume of dissolved gases which are liberated by the gasoline. From the study of the solubility of gases in gasolines, it was concluded that the volume of dissolved gases which would be liberated in the fuel line was not sufficient to cause vapor lock in a well-designed fuel-feed system. Hence, the vapor pressure of the gas-free gasoline is the only essential property of a fuel which is of interest as regards vapor lock. In this work, propane was considered as one of the liquid constituents of the fuel, but if present in more than very small amounts, sufficient gas might be liberated to cause serious interruption of flow.

In the vapor-pressure work, to which reference has been made, it was found that vapor-pressure data could be obtained from the American Society for Testing Materials distillation-data on the fuels, in all except extreme cases. With regard to the aviation fuels marketed commercially in this Country, such extreme cases are almost entirely limited to gasolines containing appreciable amounts of propane, which is an undesirable constituent of aviation fuels. Fortunately, such fuels are not very common. Accordingly, the A.S.T.M. distillation-data are adequate in the majority of cases for the practical estimation of vapor-locking temperatures at various reduced pressures or altitudes.

The vapor-locking temperature is the temperature at which the vapor pressure of the gas-free gasoline becomes equal to the external pressure on the gasoline in the fuel-feed line at the point where marked vaporization occurs. The general relation between the distillation data, the external pressure on the fuel in the feed line and vapor-locking temperature is as follows:

$$t = \frac{At_{10\text{ per cent}} + 273.1 \log\left(\frac{p}{760}\right)}{A - \log\left(\frac{p}{760}\right)} \qquad (1)$$

in which

$$A = 4.45 + 3.8 \times 10^{-2} t_{10\text{ per cent}} - 0.6\sqrt{S}$$

In this equation, t is the vapor-locking temperature in degrees centigrade, $t_{10\text{ per cent}}$ is the A.S.T.M. 10-per cent point of the gasoline also in degrees centigrade, p is the pressure on the gasoline in the fuel line expressed in millimeters of mercury and A is a constant for each gasoline dependent upon the 10-per cent point and the slope S of the A.S.T.M. curve at the 10-per cent point expressed in terms of degrees centigrade per unit percentage evaporated.

The A.S.T.M. 10-per cent point is the temperature at which 10 per cent of the gasoline is evaporated in the standard A.S.T.M. distillation-test. This temperature is read from the curve obtained by plotting the recorded temperatures against the corresponding percentages evaporated. The percentage evaporated at each point is the recorded percentage distilled plus the distillation loss. This procedure of plotting A.S.T.M. distillation-data in terms of percentages evaporated places all gasolines on a comparative basis which is not true when the customary procedure is followed of expressing distillation-data in terms of percentages distilled.

Equation (1) may also be written in terms of temperatures expressed in degrees fahrenheit and of pressures in inches of mercury.

$$t' = \frac{At'_{10\text{ per cent}} + 459.6 \log\left(\frac{p'}{29.92}\right)}{A - \log\left(\frac{p'}{29.92}\right)} \qquad (2)$$

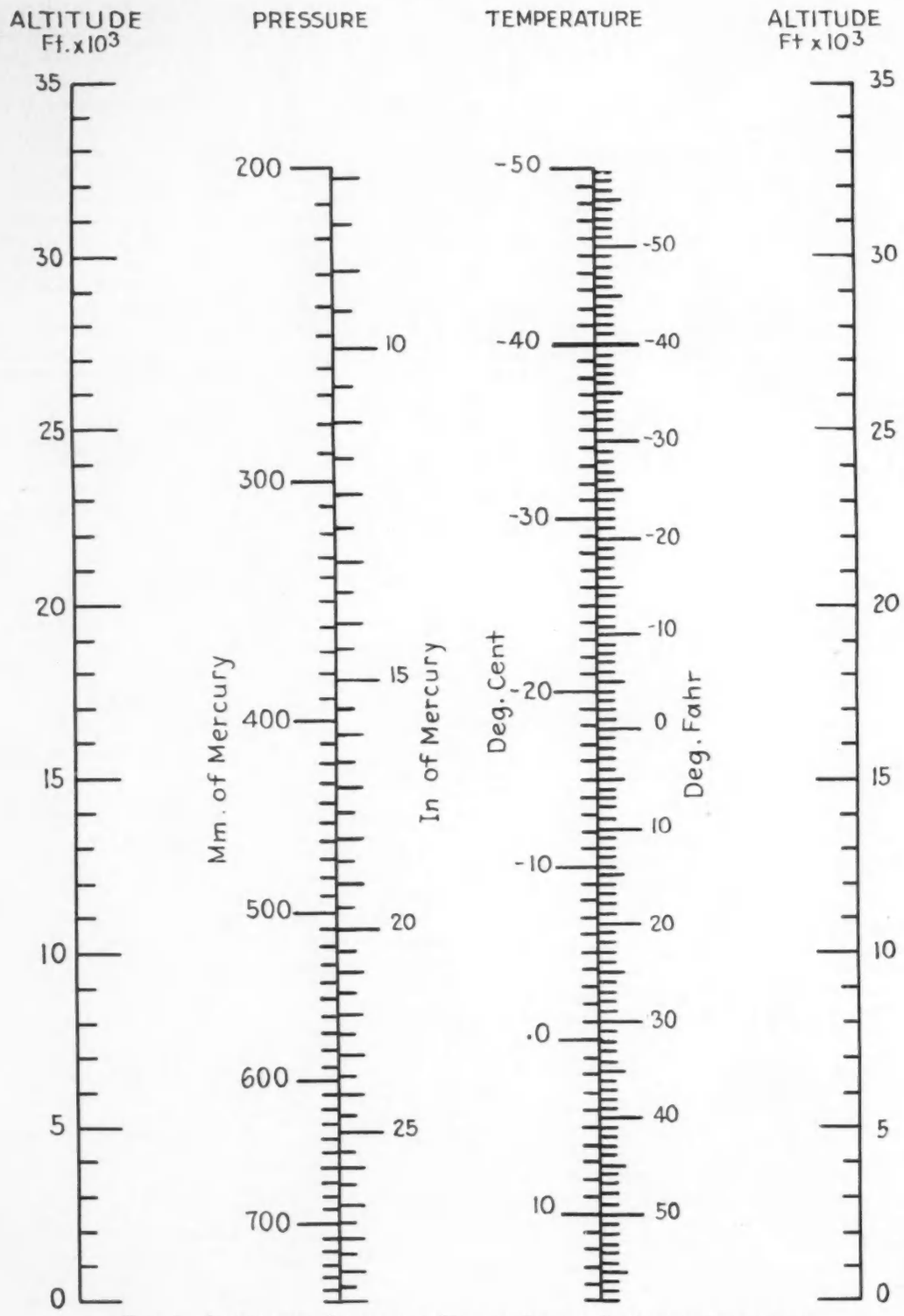


FIG. 1—ALIGNMENT CHART FOR UNITED STATES STANDARD ATMOSPHERE

where

$$A = 4.38 + 2.11 \times 10^{-3} t'_{10 \text{ per cent}} - 0.45 \sqrt{S_F}$$

In this equation, t' and $t'_{10 \text{ per cent}}$ are in degrees fahrenheit, p' is in inches of mercury and S_F is expressed in terms of degrees fahrenheit per unit percentage evaporated.

Equations (1) and (2) give accurately the temperatures at which the vapor pressure of the gas-free gasoline becomes equal to any given external pressure within the range of interest, and hence they give the predicted temperature at which vapor lock will occur at

any altitude. It should be noted that these equations do not hold as accurately for aviation fuels containing more than a few tenths of a per cent of propane. In such cases, the vapor-locking temperatures may be several degrees below those predicted from the distillation data.

In using the equations for the computation of vapor-locking temperatures at various altitudes, the pressure at any altitude can be obtained from the U. S. Standard Atmosphere table or chart[5]. A convenient graphical method of evaluating the pressure from the altitude is illustrated in Fig. 1, where for completeness the standard temperatures at various altitudes are also included. The two extreme scales represent the altitude in feet, while the two intermediate lines represent atmospheric pressure and temperature respectively. Pressures are expressed both in millimeters of mercury and in inches of mercury, while temperatures are given both in degrees centigrade and degrees fahrenheit. In reading pressures or temperatures at any altitude from the chart, connect with a straight edge the identical value of the altitude on the two extreme scales. The intersection of this line with the inner scales gives the corresponding standard pressure and temperature. This chart is, of course, subject to the errors resulting from the deviations between the standard atmosphere and the actual atmospheric conditions.

The computation of vapor-locking temperatures from the A.S.T.M. distillation-data by Equations (1) or (2) may be simplified by the use of the chart shown in Fig. 2. This chart is based on an average value of the constant A, which is taken as 4.0. The error introduced by this approximation is small. The scale on the right represents the altitude in feet; the scale on the left represents the 10-per cent point of the gasoline in degrees centigrade and degrees fahrenheit; whereas the scale in the middle represents the vapor-locking temperature of the gasoline in centigrade and fahrenheit degrees. In using the chart for evaluating the vapor-locking temperature of a given fuel at any altitude, connect by means of a straight edge the value on the scale at the right representing the chosen altitude with the value of the 10-per cent point of the fuel given on the scale at the left. The intersection of this line with the middle scale gives the vapor-locking temperature with

[5] See Bureau of Standards Miscellaneous Publication No. 82.

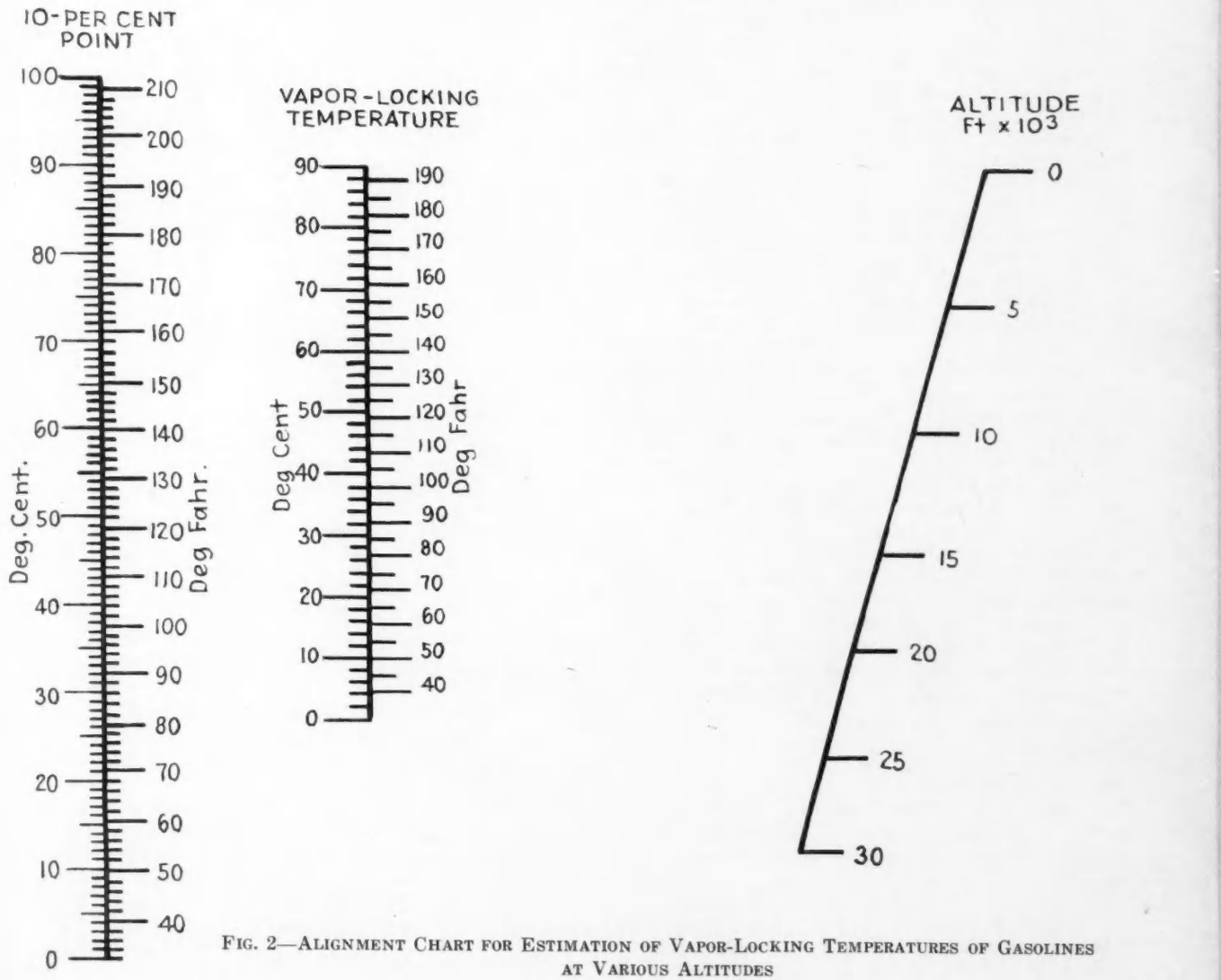


FIG. 2—ALIGNMENT CHART FOR ESTIMATION OF VAPOR-LOCKING TEMPERATURES OF GASOLINES AT VARIOUS ALTITUDES

this fuel at this altitude. For example, the vapor-locking temperature at 20,000 ft. for a fuel with a 10-per cent point of 60 deg. cent. (140 deg. fahr.) is 34 deg. cent. (93 deg. fahr.) while for one having a 10-per cent point of 40 deg. cent. (104 deg. fahr.) the temperature of the fuel at which trouble from vapor lock is predicted will be 16 deg. cent. (61 deg. fahr.). It is to be noted that the vapor-locking temperature at sea-level is the 10-per cent A.S.T.M. point. If the characteristics of the fuel system are such that the gasoline is under additional pressure or if under reduced pressure, such as occurs with a fuel pump, the atmospheric pressure at any given altitude will not truly represent the external pressure on the gasoline and correction should be made for the effects mentioned.

Airplane Fuel-Feed Systems

The fuel-feed systems used in airplanes all involve the one general principle that the fuel is fed to the carbureter under pressure. The pressure may be hydrostatic, in which case the system can be classed as "gravity feed," or the pressure may be produced mechanically, in which case the system can be designated as "pressure-feed." Pressure-feed systems ordinarily involve a reduction in pressure below that in the tank as the result of suction lift or suction produced by the rotating mechanism of the pump. Various combinations of these two types of systems can be employed satisfactorily, as illustrated by the variety of fuel systems to be found in Army, Navy and commercial airplanes.

In outlining the program for the investigation of the conditions under which vapor lock occurred in fuel-feed systems, the work was subdivided into two general phases:

(1) Study of the conditions for vapor lock in gravity-feed systems
(2) Study of the conditions for vapor lock on the suction side of the fuel pump

Investigation of vapor lock on the pressure side of the fuel pump was not included in this initial study, since the conditions for vapor lock with a gasoline under pressure can in general be computed from the measured pressure and a knowledge of the conditions for vapor lock when the gasoline is flowing under hydrostatic pressure.

The results obtained in the study of the two phases outlined are given in the following sections of this paper. No extensive results have yet been secured on the effect of variations in the detailed design of the fuel systems. It is known that variations in design can have a marked effect on the tendency to vapor lock, but it was considered advisable to obtain first the more fundamental information on the general types of systems, in order that this information would be available for the interpretation of the data on design factors.

Vapor Lock in Experimental Gravity-Feed Systems

The most important factor tending to cause vapor lock in a gravity-feed system is a constriction in the line, such as the metering orifice. Other details of construction or installation, such as arrangement of T-connections, size of tubing, siphons and so on, may have an important bearing on the vapor-locking tendency, but in general trouble from these sources can be eliminated by changes in the system. A metering orifice is, however, essential in a carbureting system and accordingly an experimental feed line was set up to study the conditions under which vapor lock occurred in the line leading to the orifice.

A schematic diagram of the fuel-feed system is shown in Fig. 3. The inner compartment of *a* was used as the reservoir for holding the gasoline during the flow experiments, and a tube was provided for reducing the pressure above the gasoline in order to simulate changes in altitude. The inner compartment was surrounded by a water bath containing a coil for the circulation of cold or hot oil to control the temperature. The intermediate compartment was in turn surrounded by a box containing ground cork for heat insulation. The gasoline flowed out of the tank through a copper tube to the orifice *d* and was heated during passage by the jacket *b* through which heated oil was circulated by

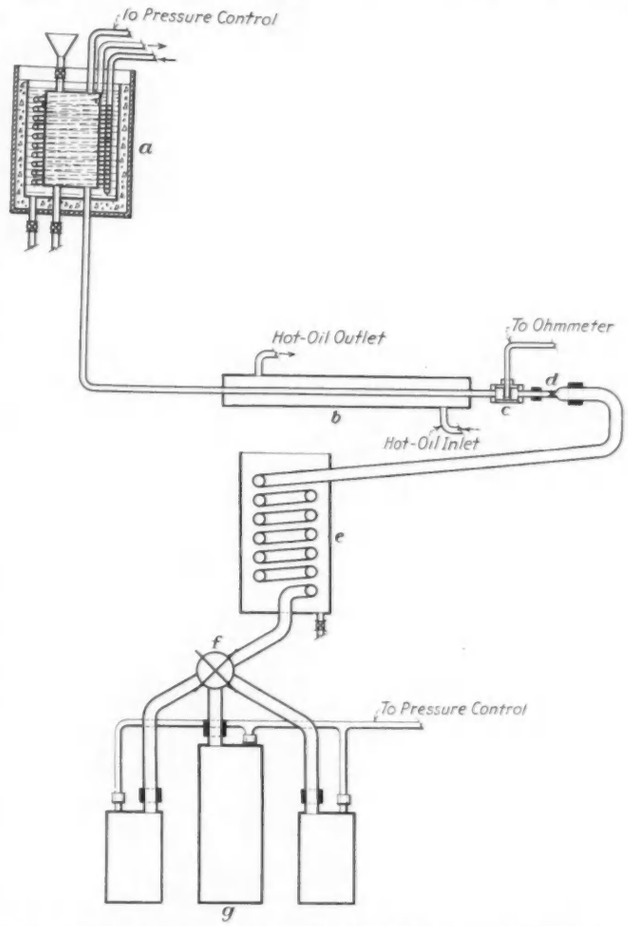


FIG. 3—DIAGRAM OF EXPERIMENTAL GRAVITY-FEED FUEL-SYSTEM

a pump. A nickel-resistance thermometer *c* served to measure the temperature of the gasoline just before it entered the orifice. After the gasoline passed through the orifice, it flowed through a large copper tube into the coil *e* which was surrounded with a freezing mixture of ice and salt. The three-way stopcock *f* was a large-bore, cork-seated valve provided with a handle for rapid manipulation. The three outlets from the valve ran to brass containers *g* fitted with tubes for reducing the pressure.

In making an experiment, gasoline was run through the funnel into the inner compartment of *a* until the

TABLE 1—DESCRIPTION OF GASOLINES AND BLENDS

Fuel	Source[6]	Remarks
104	Naturaline Co. of America	Natural Gasoline
104A		Aviation Fuel
124	Virginian Gasoline & Oil Co.	Natural Gasoline
124A		Aviation Fuel
134	Standard Oil Co. of California	Aviation Gasoline
134A		
144	Army Air Corps	1929 Supply of Domestic Aviation Gasoline
154	Standard Oil Co. of Indiana	Aviation Gasoline
184	Atlantic Refining Co.	Aviation Gasoline
194	Phillips Petroleum Co.	Natural Gasoline Aviation Fuel
204	Bureau of Standards	Blend of 50 Per Cent of Fuel 104 and 50 Per Cent of Fuel 144
314	Bureau of Standards	Propane Blend with Fuel 154 — 3.0 Per Cent Loss
324		4.5 Per Cent Loss
324A		4.0 Per Cent Loss
334		7.4 Per Cent Loss
344		1.3 Per Cent Loss
354		1.8 Per Cent Loss
364		3.2 Per Cent Loss
404	Richfield Oil Corp.	Aviation Gasoline
504	Naturaline Co. of America	Grade A Natural Gasoline
514	Bureau of Standards	Blend of 25 Per Cent of Fuel 504 and 75 Per Cent of Fuel 184
524	Bureau of Standards	Blend of 50 Per Cent of Fuel 504 and 50 Per Cent of Fuel 184

[6] The gasolines contributed by these companies are experimental samples and should not be considered as representative of their commercial products.

whole system from the three-way valve *f* to the bottom of the funnel was filled, after which the valve below the funnel was closed. The temperature of the gasoline in the tank was adjusted so that in every case it was below the temperature at which the vapor pressure of the gas-free gasoline equalled the external pressure. Oil, heated to an experimentally predetermined temperature, was circulated through the jacket *b*, the coil *e* was surrounded with the mixture of ice and salt and when everything was in readiness the three-way valve *f* was turned so that gasoline flowed into the center of the three containers *g*. The pressure in the tank *a* and in the containers *g* was the same. Temperature readings of the gasoline entering the orifice *d* were commenced, using the nickel-resistance thermometer *c* in conjunction with a direct-reading ohmmeter. When the temperature readings became constant, the three-way valve was turned so as to divert the gasoline into one of the outside containers *g*. Gasoline was allowed to flow into this container for a measured period of time, after which the valve was again closed and the weight of gasoline which flowed into the container during the measured time-interval was determined. Similar experiments were made at a number of temperatures in order to obtain a temperature-flow curve, and similar curves were obtained for other pressures and with

TABLE 2—SPECIFICATION DATA ON GASOLINES AND BLENDS

	Temperatures[7]								60 Deg./60 Deg.
	10 Per Cent		50 Per Cent		90 Per Cent		Loss, Per Cent.	Residue, Per Cent.	
Fuel	Deg. Cent.	Deg. Fahr.	Deg. Cent.	Deg. Fahr.	Deg. Cent.	Deg. Fahr.			Specific Gravity
104	45	113	65	149	109	228	1.3	1.2	0.679
104A	43	109	63	145	109	228	1.5	1.4	0.678
124	71	160	79	174	109	228	1.2	1.8	0.695
124A	69	156	76	169	109	228	0.5	1.5	0.691
134	57	135	87	189	116	241	1.1	1.1	0.718
134A	60	140	91	196	122	252	1.2	1.1	0.721
144	64	147	104	219	140	284	1.4	0.6	0.716
154	66	151	94	201	124	255	1.5	0.5	0.713
184	67	153	103	217	147	297	1.0	1.2	0.725
194	49	120	66	151	111	232	0.9	1.1	0.680
204	53	127	84	183	128	262	0.9	1.1	0.701
314	61	142	95	203	124	255	3.0	0.8	0.710
324	58	136	94	201	124	255	4.5	0.5	0.709
324A	59	138	93	199	124	255	4.0	1.2	0.710
334	43	109	93	199	124	255	7.4	0.7	0.704
344	64	147	94	201	125	257	1.3	0.7	0.713
354	64	147	94	201	124	255	1.8	0.8	0.712
364	61	142	94	201	124	255	3.2	0.8	0.710
404	62	144	100	212	138	280	2.0	1.2	0.729
504	27	81	58	136	110	230	6.2	1.2	0.671
514	47	117	92	198	140	284	1.8	1.2	0.707
524	37	99	81	178	134	273	3.8	1.2	0.696

[7] All temperatures refer to percentages evaporated.

other fuels. This general procedure was followed throughout the entire series of tests, although minor variations in procedure were used in some of the experiments, such as measuring the volume of gasoline flowing through the orifice rather than the weight. In all the experiments with the gravity system, the hydrostatic head of liquid on the jet was approximately 40 in. or equivalent to a pressure of about 1 lb. per sq. in.

One of the first series of experiments consisted of a study of the effect on the vapor-locking temperature of direction of flow through the orifice. This is illustrated in Fig. 4 for Fuel 104 at 760-mm. (29.92-in.) pressure with a flow of about 100 ml. per min. (1.585 gal. per hr.). The curve through the points represents the data very closely regardless of the direction of flow. It is to be noted how rapidly the rate of flow changes as the temperature is increased beyond a certain value. The temperature at which a marked change in rate of flow occurs is the vapor-locking temperature and it is seen how closely the predicted temperature, represented by the arrow, agrees with the observed value. It was concluded that direction of flow introduced no differences with the system employed and accordingly horizontal flow through the orifice was used in the further experiments. This conclusion was checked by experimentation with another gasoline.

A second series of measurements were made to determine the effect of volume of flow on the temperature at which vapor lock occurred. Results obtained on Fuel 134A at 760 mm. (29.92 in.) are shown in Fig. 5 for rates of flow of 95 and 660 ml. per min. (1.506 to 9.361 gal. per hr.) respectively. Within experimental error, variation in the initial rate of flow does not appear to change the vapor-locking temperature. This has been checked by experiments on two other gasolines and at pressures of 570 and 380 mm. (22.44 and 14.96 in.) of mercury. Although the vapor-locking temperature was unchanged, there did seem to be a slight tendency for the flow to be reduced more rapidly after vapor lock occurred, when employing larger volumes of flow. On the basis of these experiments, an orifice permitting a flow of about 100 ml. per min. (1.585 gal. per hr.) was used for most of the experiments to economize on gasoline consumption. It should be pointed out that variation in the rate of flow in these experiments was effected by changing the size of the orifice and not by changing the hydrostatic head of liquid or by reducing the pressure on the outlet side of the orifice. In the case of an actual system in an airplane, variation in rate of flow is controlled by the suction on the jet and the effective pressure at the metering orifice is different with every rate of flow. Such conditions are analogous to those studied experimentally in this work, in which the pressure on the outlet side of the orifice was varied.

At the completion of these introductory experiments on direction and rate of flow, a long series of results were obtained on a number of gasolines. The 100 ml. per min. (1.585 gal. per hr.) orifice was employed in a horizontal position and, in general, data were obtained at three pressures, namely, 760, 570 and 380 mm. (29.92, 22.44 and 14.96 in.), which correspond approximately to sea level, 8000 ft. and 18,000 ft. altitude. At each pressure, about 10 to 25 observations were made which was found to be sufficient to obtain a well-defined temperature-flow curve.

Flow data were obtained on 22 fuels, of which 7 were blends containing various amounts of propane. A de-

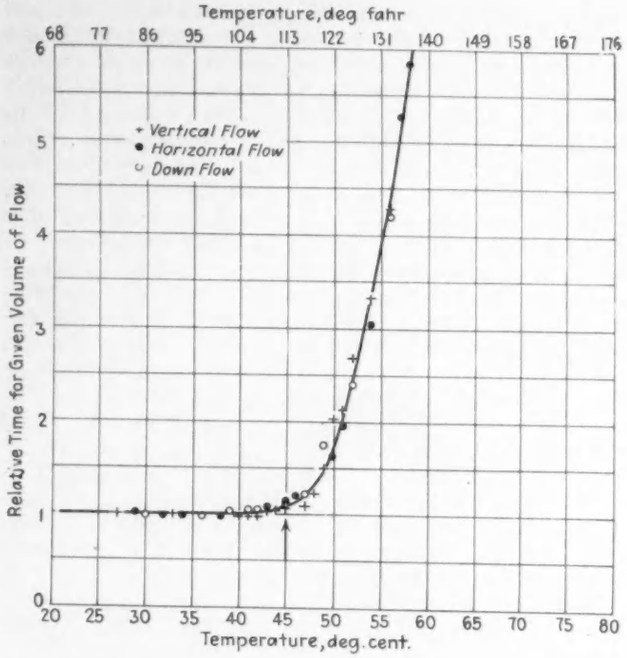

Temperature, deg fahr
68 77 86 95 104 113 122 131 140 149 158 167 176
Relative Time for Given Volume of Flow
+ Vertical Flow
• Horizontal Flow
○ Down Flow
20 25 30 35 40 45 50 55 60 65 70 75 80
Temperature, deg. cent.

FIG. 4—TEMPERATURE-FLOW CURVE FOR FUEL 104 ILLUSTRATING NEGLIGIBLE EFFECT OF DIRECTION OF FLOW THROUGH THE ORIFICE

scription of these fuels is given in Table 1 and specification data are given in Table 2. While the aviation gasolines were not purchased in the open market, it is believed that the general conclusions drawn from the data on these samples are applicable to commercial aviation gasolines. In recording the specification data in Table 2, all temperatures refer to the various percentages evaporated. These temperatures were obtained by plotting the A.S.T.M. distillation-curves with the loss at the front end.

Representative temperature-flow curves at three pressures are shown in Figs. 6, 7 and 8 for fuels 104A, 124 and 144 respectively. For comparison, the A.S.T.M. curves of these fuels are plotted at the top of each fig-

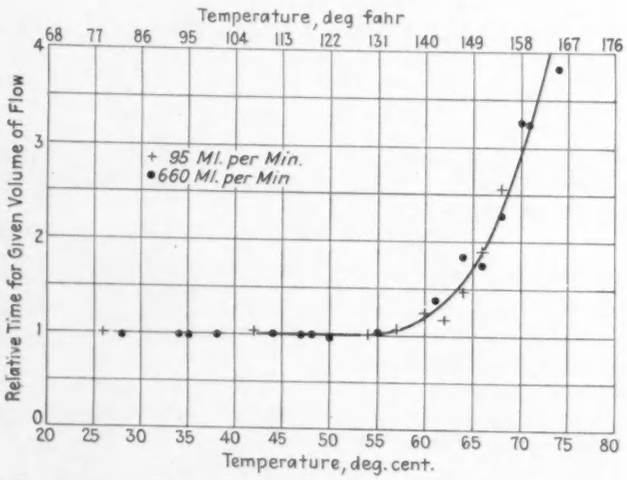

Temperature, deg fahr
68 77 86 95 104 113 122 131 140 149 158 167 176
Relative Time for Given Volume of Flow
+ 95 Ml. per Min.
• 660 Ml. per Min
20 25 30 35 40 45 50 55 60 65 70 75 80
Temperature, deg. cent.

FIG. 5—TEMPERATURE-FLOW CURVE FOR FUEL 134A ILLUSTRATING NEGLIGIBLE EFFECT OF VELOCITY OF FLOW THROUGH FUEL-FEED LINE

TABLE 3—TEMPERATURE-FLOW DATA ON GASOLINES AND NON-PROPANE BLENDS

Pressure—760 Mm. (29.92 In.) of Mercury, Corresponding to Sea Level — 570 Mm. (22.44 In.) of Mercury, Corresponding to 8,000-Ft. Altitude — 380 Mm. (14.96 In.) of Mercury, Corresponding to 18,000-Ft. Altitude

Fuel	Predicted Vapor-Locking Temperature	Temperatures Corresponding to Various Percentage Decreases in Rate of Flow: 10 Per Cent.	15 Per Cent.	20 Per Cent.	50 Per Cent.	Predicted Vapor-Locking Temperature	Temperatures Corresponding to Various Percentage Decreases in Rate of Flow: 5 Per Cent.	10 Per Cent.	20 Per Cent.	50 Per Cent.	Predicted Vapor-Locking Temperature	Temperatures Corresponding to Various Percentages Decrease in Rate of Flow: 0 Per Cent.	5 Per Cent.	10 Per Cent.	50 Per Cent.
	Deg. Cent.	Deg. Fahr.	Deg. Cent.	Deg. Fahr.	Deg. Cent.	Deg. Fahr.	Deg. Cent.	Deg. Fahr.	Deg. Cent.	Deg. Fahr.	Deg. Cent.	Deg. Fahr.	Deg. Cent.	Deg. Fahr.	Deg. Cent.	Deg. Fahr.
104	45	113	44.5	112	45	113	46	115	48.5	119	36	97	35	95	36	97	37.5	100	39	102	23.5	74	22	72	24.5	72	25.5	78	29	84
104A	43	109	42	108	43	109	43.5	110	47	117	34	93	34	93	35	95	36.5	98	33	102	21.5	71	22	72	25	77	26	79	29	84
124	71	160	71	160	72	162	72.5	163	75.5	168	61.5	143	61.5	143	62.5	144	64	147	67	153	49	120	49	120	50.5	123	51.5	125	54.5	130
124A	69	156	65.5	150	67	153	68	154	71	160	59	138	56	133	57.5	135	59	138	62	144	47	117	40	104	43	103	45	113	50	122
134	57	135	55.5	132	56.5	134	57.5	136	61.5	143	47	117	45	113	47	117	49	120	52	126	34	93	34	93	35.5	96	38	100	41.5	107
134A	60	140	59	138	60	140	60.5	141	64	147	50	122	49.5	121	50.5	123	52	126	54.5	130	36	97	35	95	38	100	39	102	43	109
144	64	147	64	147	65	149	68	151	70	158	53.5	128	50	122	51	124	54	129	58	133	40	104	35	95	38.5	101	40	104	47	117
154	66	151	62.5	145	64	147	65.5	150	70	158	56	133	55.5	132	56.5	134	58	135	61	142	42	108	39	102	41.5	107	43	109	48	118
184	67	153	66	151	67.5	154	69	156	73	163	56.5	134	55	131	58.5	136	58.5	137	62	144	43	109	41	105	44	111	45	113	50	122
194	49	120	48	118	49	120	49.5	121	52	126	39.5	103	38.5	101	40	104	41.5	107	43.5	110	27	81	27	81	29	84	30.5	87	34	93
204	53	127	53.5	128	54.5	130	55	131	58	136	43	109	44	111	46	115	48	118	50	122	30	86	33	85	34	92	36	97	41	106
504	27	81	25.5	78	26.5	80	27	81	28.5	83	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...	...
514	47	117	46	115	47	117	48	118	52	126	37	99	37	99	39	102	41	106	44	111	24	75	23	73	26	79	27	81	33	91
524	37	99	38	100	39	102	40	104	43	109	27	81	28	82	29.5	85	31.5	89	35	95	15	59	14	57	17	63	18.5	65	24	75

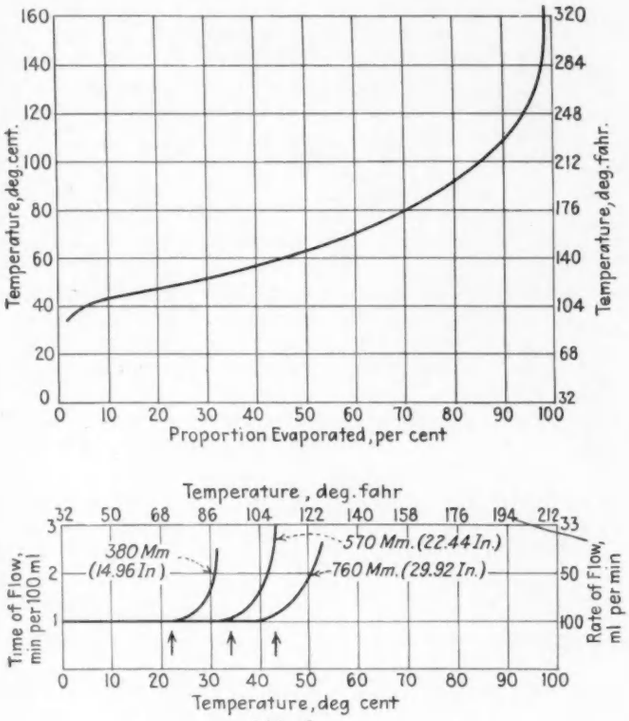

Temperature, deg. cent.
Temperature, deg. fahr.
Proportion Evaporated, per cent
Temperature, deg. fahr
Time of Flow, min. per 100 ml
Rate of Flow, ml per min
380 Mm (14.96 In)
570 Mm. (22.44 In.)
760 Mm. (29.92 In.)
Temperature, deg cent

FIG. 6

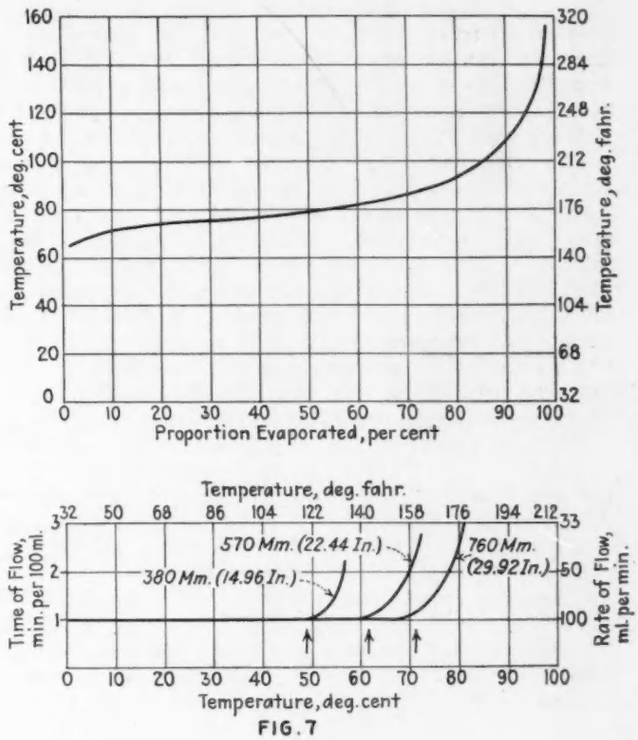

Temperature, deg. cent
Temperature, deg. fahr.
Proportion Evaporated, per cent
Temperature, deg. fahr.
Time of Flow, min. per 100 ml.
Rate of Flow, ml. per min.
570 Mm. (22.44 In.)
760 Mm. (29.92 In.)
380 Mm. (14.96 In.)
Temperature, deg. cent

FIG. 7

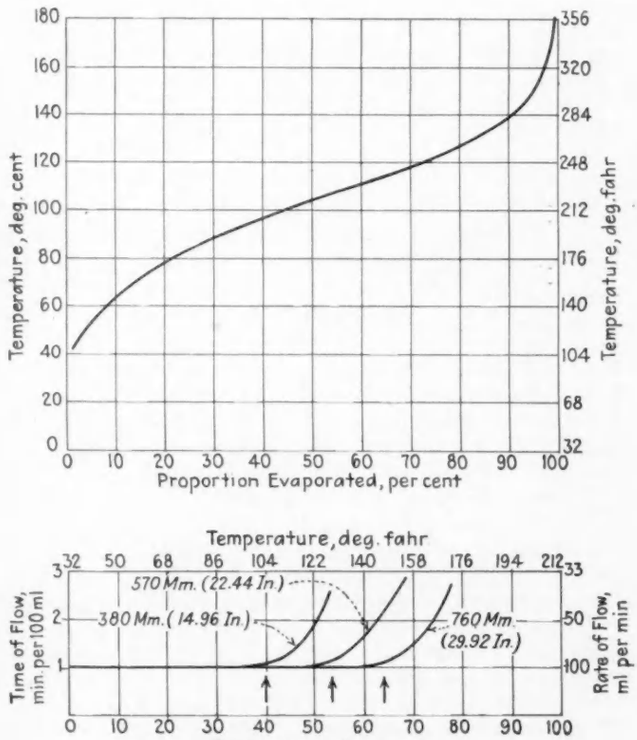

Temperature, deg. cent
Temperature, deg. fahr
Proportion Evaporated, per cent
Temperature, deg. fahr
Time of Flow, min. per 100 ml
Rate of Flow, ml per min
570 Mm. (22.44 In.)
380 Mm. (14.96 In.)
760 Mm (29.92 In.)
Temperature, deg. cent

FIG. 8

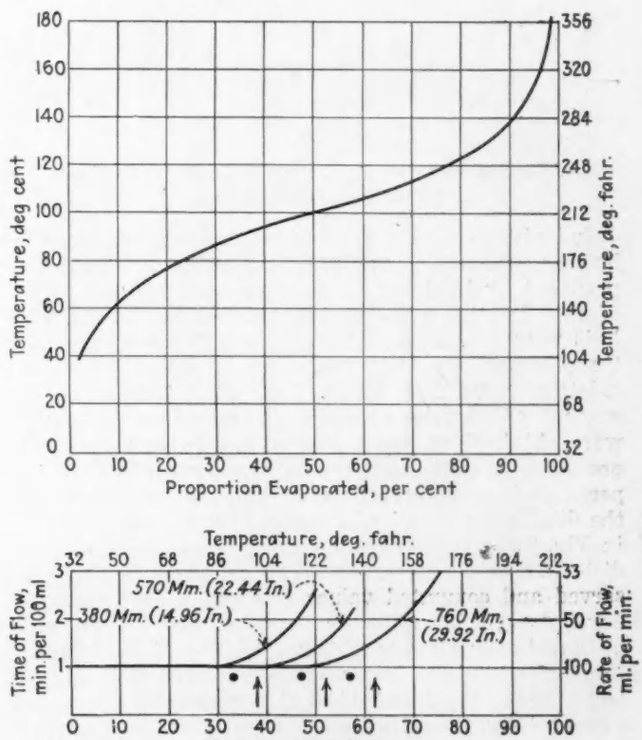

Temperature, deg. cent
Temperature, deg. fahr.
Proportion Evaporated, per cent
Temperature, deg. fahr.
Time of Flow, min. per 100 ml
Rate of Flow, ml. per min.
570 Mm. (22.44 In.)
380 Mm. (14.96 In.)
760 Mm. (29.92 In.)
Temperature, deg. cent.

FIG. 9

Fig. 6—Fuel 104A
Fig. 8—Fuel 144

Fig. 7—Fuel 124
Fig. 9—Fuel 404

A. S. T. M. Distillation-Curves and Temperature-Flow Curves at Various Pressures for Different Fuels

ure. The arrows at the bottom of each set of temperature-flow curves represent the predicted vapor-locking temperatures computed by Equation (1). It is seen that they are in very good agreement with the temperatures at which the rate of flow begins to change rapidly.

With each fuel, the temperatures at which the rate of flow decreased by various percentages were read from the temperature-flow curves. These data are given in Table 3 for the 14 gasolines and blends which did not contain propane. For comparison, the predicted vapor-locking temperatures computed by Equation (1) are also included. There is good agreement between the predicted values and the temperatures at which the rate of flow has decreased by about 10 per cent, particularly at an atmospheric pressure of 760 mm. (29.92 in.). A variation in flow by this amount is within the normal fluctuations in mixture ratio and will not seriously affect engine operation. Attention is called to the small temperature-change required to decrease the flow by 50 per cent, which would certainly cause engine stoppage. At the lower pressures, there is a tendency for the predicted values to show better agreement with the temperatures at which the flow has decreased by amounts smaller than 10 per cent, which is on the safe side. It should be emphasized that the predicted vapor-locking temperatures are computed entirely from the A.S.T.M. distillation-data.

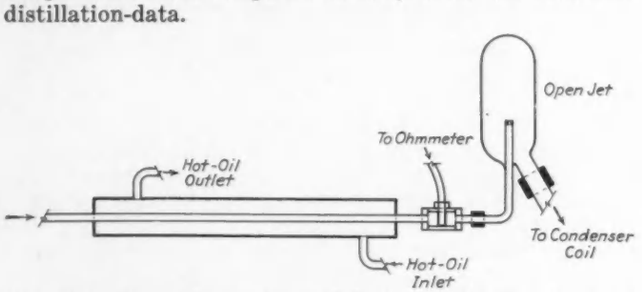


FIG. 10—DIAGRAM OF EXPERIMENTAL OPEN JET USED IN GRAVITY-FEED SYSTEM

In order to study the effect of propane on the vapor-locking tendency of gasolines, a number of blends were prepared by bubbling propane gas for various periods of time through respective samples of Fuel 154. Since it is very difficult to determine the propane content of these blends, the percentage loss in the A.S.T.M. distillation is used in Table 1 as an indication of the amount of propane present. Temperature-flow curves were obtained on these blends, mostly at atmospheric pressure. In every case, the observed vapor-locking temperature is considerably lower than that predicted from the distillation data by Equation (1). This is illustrated in Fig. 9 for Fuel 404, the arrows being the values predicted from the equation. The deviation between observed and computed values is about 8 deg. cent. (14 deg. fahr.). The vapor pressures of this fuel were determined over a range of temperature and the temperatures at which the vapor pressure is 760, 570 and 380 mm. (29.92, 22.44 and 14.96 in.) respectively are indicated in Fig. 9 by the solid circles. The agreement is good between the observed vapor-locking temperatures and those predicted from the vapor-pressure measurements. The lack of agreement in the case of values from Equation (1) is due to the fact, pointed out previously, that this equation will not satisfactorily reproduce the experimental vapor-pressure data on propane blends. A summary of the data obtained on the eight fuels known to contain propane is given in Table 4. It is seen that the observed vapor-locking temperatures are considerably lower than the values predicted from the A.S.T.M. distillation-data. Vapor-pressure data had been obtained on only one of these fuels, namely Fuel 404, but the vapor pressure of similar propane blends with Fuel 154 had been determined. From these vapor-pressure data, the vapor-locking temperatures were estimated and are given in the fourth column of the table. The agreement is fair between the values estimated from vapor pressures and the observed temperatures at which the rate of flow begins to decrease materially. It is interesting to note that the change in the rate of flow with temperature after vapor lock occurs is much less in the case of the propane blends than in the case of the fuels listed in Table 3.

Most of the propane blends studied showed A.S.T.M. distillation-losses higher than that permitted in present Federal Specifications for aviation gasolines, which is fixed at a maximum value of 2 per cent. However, those with distillation losses of 2 per cent or less showed a marked deviation between the observed vapor-locking temperatures and those computed from Equation (1). The vapor-locking temperature of the original gasoline

TABLE 4—TEMPERATURE-FLOW DATA ON PROPANE BLENDS

Fuel	A.S.T.M. Loss, Per Cent.	A.S.T.M. Data	Predicted Vapor-Locking Temperatures from Vapor-Pressure Data[a]		Temperatures Corresponding to Various Percentage Decreases in Rate of Flow: 5 Per Cent.		10 Per Cent.		20 Per Cent.		50 Per Cent.	
		Deg. Cent.	Deg. Fahr.	Deg. Cent.	Deg. Fahr.	Deg. Cent.	Deg. Fahr.	Deg. Cent.	Deg. Fahr.	Deg. Cent.	Deg. Fahr.	Deg. Cent.	Deg. Fahr.
Pressure—760 Mm. (29.92 In.) of Mercury Corresponding to Sea Level.													
314	3.0	61	142	47	117	48	118	50	122	52.5	127	57.5	136
324	4.5	58	136	43	109	38	100	40	104	44	111	51	124
324A	4.0	59	138	45	113	44	111	46	115	49	120	56	133
334	7.4	43	109	26	79	24	75	27	81	30	86	37	99
344	1.3	64	147	59	138	57	135	58	136	61	142	65	149
354	1.8	64	147	54	129	51	124	53	127	55	131	60	140
364	3.2	61	142	47	117	47	117	49	120	52	126	57	135
404	2.0	62	144	57	135	51	124	52.5	127	55.5	132	61.5	143
Pressure—570 Mm. (22.44 In.) of Mercury Corresponding to an Altitude of 8,000 Ft.													
314	3.0	50	122	36	97	40	104	41	106	43.5	110	49	120
404	2.0	52	126	47	117	39	102	42	108	45.5	114	52	126
Pressure—380 Mm. (14.96 In.) of Mercury Corresponding to an Altitude of 18,000 Ft.													
354	1.8	40	104	30	86	33	91	34	93	36	97	40	104
404	2.0	38	100	33	91	32.5	91	34	93	36	97	40.5	105

[a] With the exception of the data on Fuel 404, the vapor-locking temperatures in this column were estimated from vapor-pressure data on similar propane blends with the same Fuel 154.

154 is 66 deg. cent. (151 deg. fahr.) at 760-mm. (29.92-in.) pressure, and it is seen that the addition of propane to this fuel lowers the vapor-locking temperature markedly, in the most extreme case (Fuel 334) by about 40 deg. cent. (72 deg. fahr.). This indicates the undesirability, from the standpoint of vapor lock, of having propane in an aviation gasoline.

In addition to the flow experiments using the submerged orifice, some measurements were made with an open jet. A schematic diagram of the open jet is shown in Fig. 10, the remainder of the apparatus being the same as that shown in Fig. 3. The results obtained with this type of jet are illustrated in Fig. 11 for Fuel 124A, where comparison is made with the values obtained using the submerged jet. The data with the two types of jet are seen to be in good agreement and this has been checked by measurements with another gasoline.

The general conclusion from the flow experiments with a gravity-feed system is that the A.S.T.M. distillation-data are adequate to give the conditions under which vapor lock occurs in such a system when using

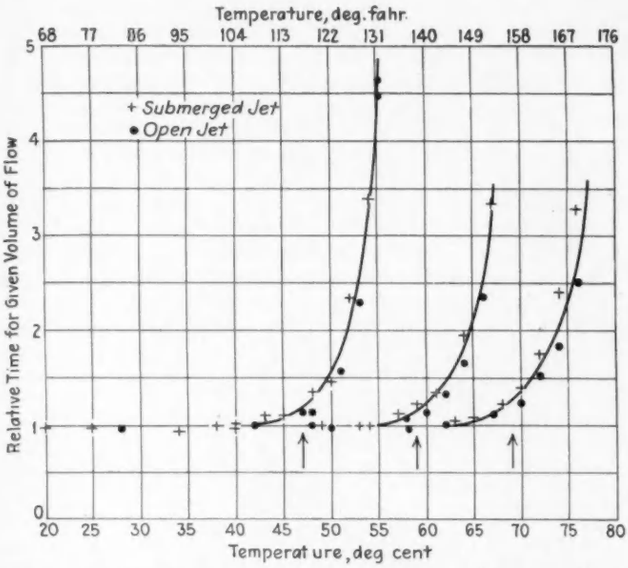

Temperature, deg. fahr.
68 77 86 95 104 113 122 131 140 149 158 167 176
+ Submerged Jet
• Open Jet
Relative Time for Given Volume of Flow
0 1 2 3 4 5
20 25 30 35 40 45 50 55 60 65 70 75 80
Temperature, deg cent

FIG. 11—TEMPERATURE-FLOW CURVES AT VARIOUS PRESSURES FOR FUEL 124A COMPARING RESULTS WITH OPEN AND SUBMERGED JETS

fuels free from propane. If the gasoline contains more than traces of propane, vapor lock may occur at temperatures considerably lower than those predicted from the A.S.T.M. data. Due to the marked effect of propane on the tendency to vapor lock, an aviation gasoline should be as free from propane as possible.

The experiments outlined have been conducted in gravity-feed systems which might be considered as ideal in that the orifice was essentially the only part of the system which induced bubble formation. The line leading to the orifice could, however, be designed so that the volume of flow would be materially different at different pressures, without affecting the temperature at which the relative rate of flow would change rapidly at any one pressure. For example, in one set of experiments the flow through the system was about 620 ml. per min. at

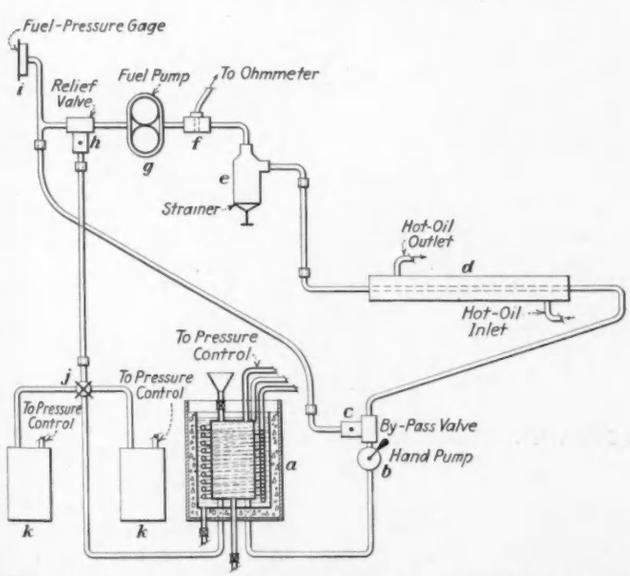


FIG. 12—DIAGRAM OF EXPERIMENTAL SYSTEM USING FUEL-FEED PUMP

TABLE 5—TEMPERATURES OF COMPLETE INTERRUPTION OF FLOW

Fuel	Suction Lift, Ft.	Pump Speed, R.P.M.	Pressure		Temperature of Complete Interruption		Predicted Temperature	
			Mm.	In.	Deg. Cent.	Deg. Fahr.	Deg. Cent.	Deg. Fahr.
184	0	700	760	29.92	65	149	67	153
			570	22.44	48	118	56	133
			380	14.96	19	66	43	109
		1,200	760	29.92	65	149	67	153
			570	22.44	53	127	56	133
			380	14.96	30	86	43	109
		1,770	760	29.92	65	149	67	153
			570	22.44	54	129	56	133
			380	14.96	33	91	43	109
	2	700	760	29.92	63	145	65	149
			570	22.44	42	108	54	129
		1,200	760	29.92	63	145	65	149
			570	22.44	44	111	54	129
			380	14.96	17	63	40	104
		1,770	760	29.92	63	145	65	149
			570	22.44	46	115	54	129
			380	14.96	19	66	40	104
	4	700	760	29.92	62	144	64	147
			570	22.44	35	95	52	126
		1,200	760	29.92	62	144	64	147
			570	22.44	39	102	52	126
		1,770	760	29.92	62	144	64	147
			570	22.44	42	108	52	126
			380	14.96	16	61	37	99
	6	700	760	29.92	61	142	62	144
			570	22.44	26	79	50	122
		1,200	760	29.92	61	142	62	144
			570	22.44	31	88	50	122
		1,770	760	29.92	61	142	62	144
			570	22.44	38	100	50	122
194	0	700	760	29.92	47	117	49	120
			570	22.44	32	90	39	102
		1,200	760	29.92	47	117	49	120
			570	22.44	35	95	39	102
			380	14.96	18	64	27	81
		1,770	760	29.92	47	117	49	120
			570	22.44	36	97	39	102
			380	14.96	20	68	27	81
	2	700	760	29.92	45	113	47	117
			570	22.44	21	70	37	99
		1,200	760	29.92	45	113	47	117
			570	22.44	29	84	37	99
		1,770	760	29.92	45	113	47	117
			570	22.44	32	90	37	99
	4	700	760	29.92	44	111	46	115
		1,200	760	29.92	44	111	46	115
			570	22.44	25	77	35	95
		1,770	760	29.92	44	111	46	115
			570	22.44	29	84	35	95
	6	700	760	29.92	43	109	44	111
		1,200	760	29.92	43	109	44	111
			570	22.44	20	68	33	91
		1,770	760	29.92	43	109	44	111
			570	22.44	25	77	33	91

pressures ranging from 760 to 380 mm. (29.92 to 14.96 in.) of mercury. Insertion of two elbows in the line near the tank changed the flow before vapor lock occurred to the following values: 760 mm. (29.92 in.)—620 ml. per min. (9.827 gal. per hr.); 570 mm. (22.44 in.)—480 ml. per min. (7.508 gal. per hr.); 380 mm. (14.96 in.)—240 ml. per mm. (4.804 gal. per hr.). In these experiments, the same orifice was used, and the marked change in flow appears to have been caused by trapping of bubbles of gas or vapor which, under a wide variety of conditions, pass back through the line into the tank against the flow of gasoline. Even with this change in flow, the temperatures at which the relative rates of flow began to change rapidly were the same as in the experiments with the original system where the total volume of flow was practically unaffected by pressure until vapor lock occurred. Similar phenomena have been observed when the size of the feed line was decreased, although in extreme cases of this nature the vapor-locking temperatures may be lowered. Further

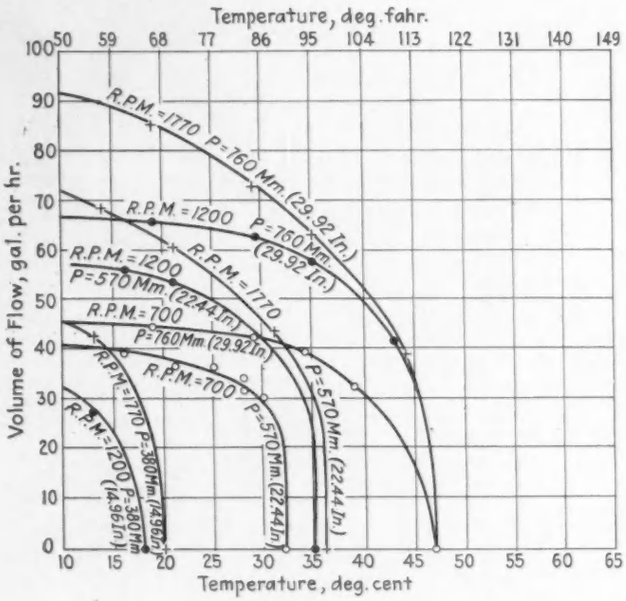


FIG. 13

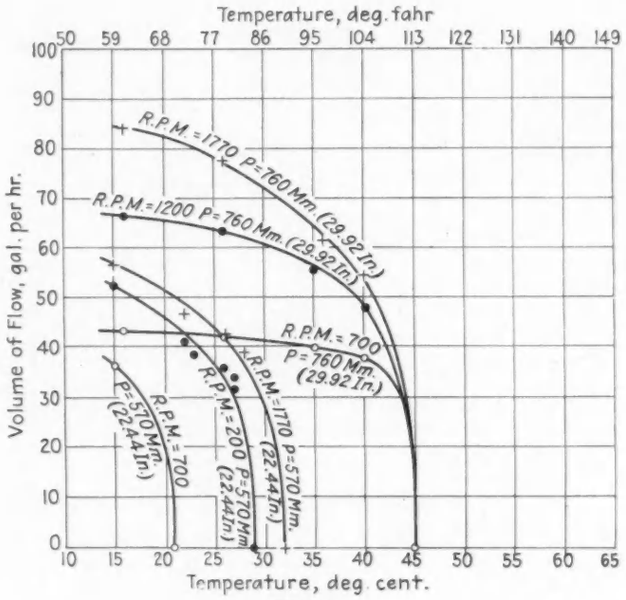


FIG. 14

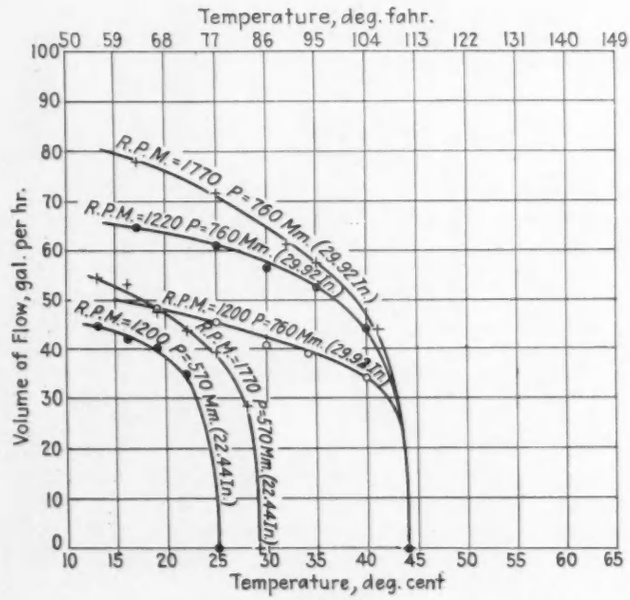


FIG. 15

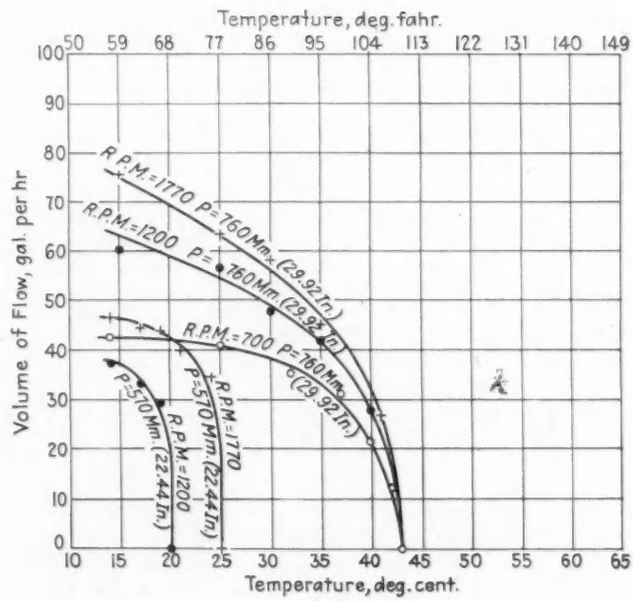


FIG. 16

Fig. 13—0 Ft.
Fig. 15—4 Ft

Fig. 14—2 Ft.
Fig. 16—6 Ft.

TEMPERATURE-VOLUME CURVES FOR FUEL 194 WITH DIFFERENT SUCTION-LIFTS

work on these design factors is under way and the results will be embodied in a subsequent paper.

Vapor Lock on the Suction Side of the Fuel Pump

For a study of the conditions under which vapor lock occurs on the suction side of the fuel pump, an experimental system was set up employing a standard gear fuel-pump. A diagram of this system is shown in Fig. 12. The tank *a* is the same as was used in the gravity-feed system with the exception of an additional inlet at the bottom for the return line from the pump. The gasoline flows through a hand pump *b* to a bypass valve *c* and thence through the jacket *d*, through which hot oil could be circulated. The additional lead from the bypass valve was for the purpose of priming the gear pump before each experiment. From the oil jacket, the gasoline passed through a strainer *e*, around the bulb of the nickel-resistance thermometer *f* and into the gear pump *g*. On the pressure side of the pump, the gasoline was led to a relief valve *h*, connected with a pressure gage *i*. When the pressure built up in the relief valve, the return line to the tank was automatically opened. This return line passed through the three-way cork-seated valve *j*, and from there either to the tank or to one of the containers *k*, for measuring the volume of flow. Gasoline only flowed through the line between the bypass valve *c* and the relief valve *h*, when the gear pump was being primed.

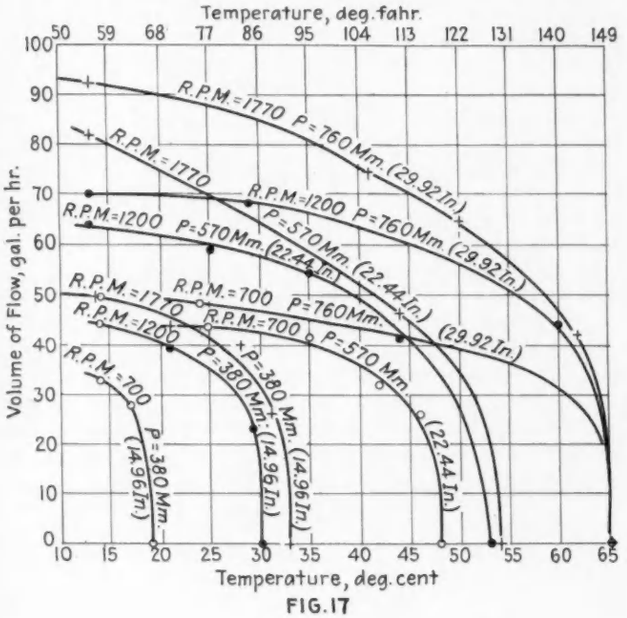


FIG. 17

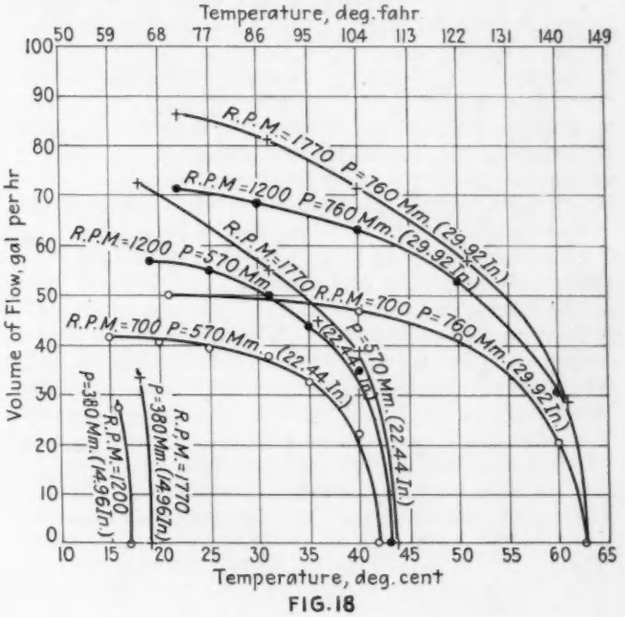


FIG. 18

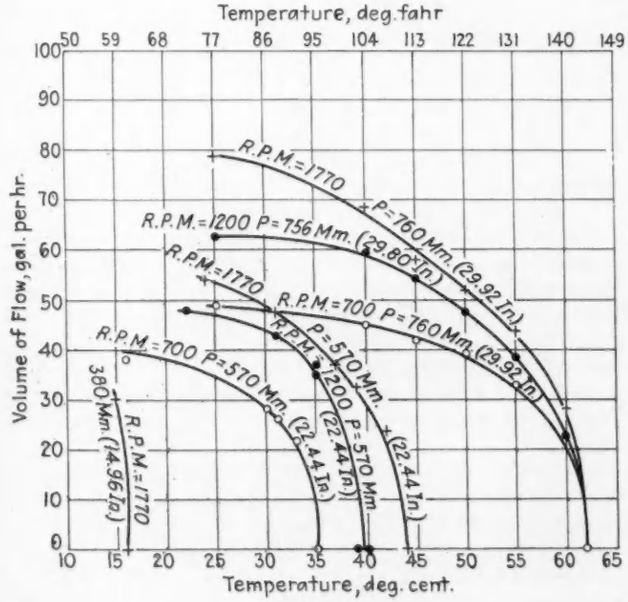


FIG. 19

Fig. 17—0 Ft.

Fig. 19—4 Ft.

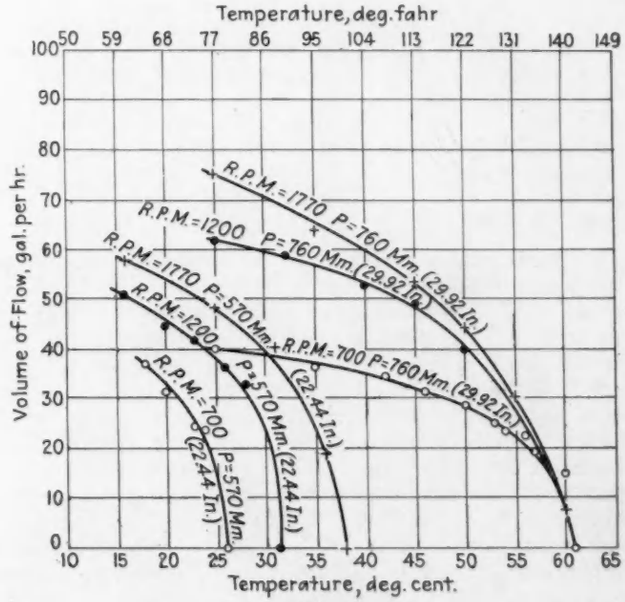


FIG. 20

Fig. 18—2 Ft.

Fig. 20—6 Ft.

TEMPERATURE-VOLUME CURVES FOR FUEL 184 WITH DIFFERENT SUCTION-LIFTS

Provision was made for regulating the speed of the pump and for adjusting the height of suction lift. Three pump speeds were used, namely, 700, 1200 and 1770 r.p.m., and four heights of suction lift were employed, namely, 0, 2, 4 and 6 ft. With each speed and each height, measurements were made with two gasolines at the two pressures, 760 and 570 mm. (29.92 and 22.44 in.) and also at 380 mm. (14.96 in.) in those cases where it was possible to lift the gasoline. Under each set of conditions, the flow was measured at a series of temperatures. The experimental system shown in Fig. 12 differs from that employed in actual systems, in that the fuel bypassed by the relief valve was measured, whereas in actual systems the main fuel-flow is direct from the relief valve to the carbureter. The results obtained, however, should not differ materially in the two cases, since in the system employed, all of the flow is bypassed and, when the pressure dropped so low that none was bypassed, the volume of flow in an actual system under the same conditions would be very much too small to permit engine operation. In these experiments, the gasoline in the tank was changed frequently in order to avoid errors caused by evaporation loss.

The results on Fuel 194 are shown in Figs. 13 to 16 for suction lifts of 0, 2, 4 and 6 ft. respectively, while similar data on Fuel 184 are shown in Figs. 17 to 20. In the figures, the total volume of gasoline pumped, expressed in gallons per hour, is plotted against the corresponding temperature for each set of conditions. Certain general comments can be made regarding these curves:

(1) In every case, the volume of gasoline pumped decreases rapidly as the temperature of the gasoline is increased.

(2) The volume of gasoline pumped at a constant temperature is, in general, not in proportion to the speed of the pump, although such a relation is approached as the temperature decreases.

(3) At the higher temperatures, the volume of gasoline pumped at atmospheric pressure tends to be independent of the speed, so long as the speed is above a certain critical value, which is below 700 r.p.m. for the conditions mentioned.

(4) As the pressure on the gasoline is reduced, the curves at different speeds tend to separate by amounts dependent upon the pressure and the height of suction lift. This is probably due to variations in the suction of the pump with change in speed and to the amount of vapor or gas which the pump can handle without stopping the flow.

The temperatures at which the flow is completely interrupted are given in Table 5 for the various sets of conditions. The predicted temperatures for complete interruption of flow are computed from the distillation data and are corrected for the reduction in pressure by the column of liquid being lifted. At atmospheric pressure, the agreement between observed and predicted values is good. At all pressures, however, there is a tendency for the observed values to be lower which is presumably due to the suction exerted by the rotating mechanism of the pump and to the fact that the pump can handle only certain volumes of vapor at each speed.

The general conclusion from this study of the efficiency of a gear pump operated under various conditions is that the vapor-locking temperatures are in general lower than the values which would be predicted from vapor-pressure data, and in some cases the differences are very marked. If the fuel system requires the use of a fuel pump, it appears evident from the data given that the pump should be situated as nearly as possible on a level with the tank, so as to have a minimum suction-lift. A lowering of the pump level from 6 to 2 ft., for example, in the case of the fuels studied, would be equivalent to raising the ceiling of the plane many thousands of feet and would materially increase the safety of operation.

Acknowledgment

This investigation was undertaken as part of a fundamental study of vapor lock in airplane fuel-systems under the direction of the Research Committee of this Society and was made possible by a contribution from the Naturaline Co. of America. The supervision of this investigation was accepted by the Cooperative Fuel Research Steering Committee in September, 1929, due to the vital importance of the vapor-lock problem to the petroleum, automotive and aircraft industries.

THE DISCUSSION

S. TIJMSTRA[9]:—I note that Dr. Bridgeman relies on the vapor pressure to obtain an indication of the tendency of a gasoline to vapor lock. I think we all agree on that in principle. He also found a correlation between vapor pressure and the 10-per cent point of the A.S.T.M. distillation. What kind of vapor pressure does he think is important in connection with vapor lock? At our laboratory we have done considerable work on this and found that the true vapor-pressure, that is to say, with a zero vapor-volume, is not a proper indication of vapor lock. We found that the fuel system can take care of a certain volume of vapors without vapor lock occurring. If we want to connect gasification trouble with vapor pressure we have to use a vapor-pressure measured under conditions of some 10 to 20 volumes of vapor. No simple relation exists between this vapor pressure and the true vapor-pressure. Fuels having the same vapor-pressure may have entirely different vapor-pressures if measured under conditions of some 10 to 20 volumes of vapor. Therefore, if vapor pressure is to be used, it should be determined under the same conditions as vapor lock occurs in the fuel system.

O. C. BRIDGEMAN:—We can design a fuel system which will require a certain type of vapor-pressure test. We can design a series of fuel systems, each of which will require a different kind of vapor-pressure test to get a correlation. My main thesis from this work is that in such cases the fuel system can be redesigned so that the vapor-locking temperature will be given by the so-called true vapor-pressure, rather than by some other test.

[9] Assistant to the vice-president of manufacturing, Shell Petroleum Corp., St. Louis.

Correlation between Vapor Pressure and the 10-Per Cent Point

MR. TIJMSTRA:—I still believe that a vapor-pressure method where the determination is made under conditions of a fairly high vapor-liquid ratio will give a better indication of a tendency to vapor lock. I further believe that if we want to study the relation between gas-lock tronble and vapor pressure we should not use the 10-per cent point of the A.S.T.M. distillation-curve as an indirect method to determine the vapor pressure. The question whether a vapor pressure can be deducted from the A.S.T.M. distillation-curve is another matter. If we discuss vapor lock we should try to find a relation between vapor lock and vapor pressure under conditions as they occur in the fuel system.

Mr. Bridgeman discusses in his paper mainly what occurs in the jet of the carbureter. Manufacturers of aviation engines tell me that they have very little trouble with gas lock in the jet, the trouble occurring in the suction side of the fuel pump. Practically all aviation-engine fuel-systems are equipped with fuel pumps, so we should look to the suction side of the pump to find the cause of the gas-lock trouble.

MR. BRIDGEMAN:—We studied a large number of aviation and automobile-engine fuels and found that in the case of those fuels which did not contain propane the percentage change in vapor pressure with a given increase in the volume of vapor space was about the same for all of the fuels. In other words, a consistent change in vapor pressure was found which did not change the order of the gasolines. In the case of gasolines containing propane the order was changed, however, in some cases. If we are talking about vapor lock in automobile fuel-systems or with automobile fuels, then the 10-per cent point may not be as good an indication of vapor lock as in the case of aviation fuels, but in this particular discussion I was limiting myself to aviation fuels. My understanding is that aviation fuels containing propane are comparatively rare. Under those conditions the relation we found between the vapor pressure and the 10-per cent point on the A.S.T.M. distillation-curve was sufficient justification for taking the 10-per cent point as an indication of the predicted condition for vapor lock, which seems to have been borne out in the gravity-feed experiments. Of course in the fuel-pump experiments the observed conditions for vapor lock are lower than the predicted, due to induced suction by the pump. In the case of automobile fuel-systems the problem is much more complicated because not only have we greater diversity of fuels but we also may have complications arising from the use of a vacuum tank with a line leading to the manifold, whereby the gasoline tends to stabilize in the pump and thereby change its vapor-locking characteristics.

MR. TIJMSTRA:—I saw an article by G. G. Oberfell[10] a few weeks ago containing a correlation between Reid vapor-pressure results and corrected 10-per cent A.S.T.M. distillation-point, and the correlation was not so very good. I noticed particularly on Naturaline, an aviation fuel, that the 10-per cent point and the vapor pressure did not check at all. It does not contain any propane because it is carefully fractionated, but I thought the fairly large pentane content was the cause of the failure of the correlation to check through.

MR. BRIDGEMAN:—Our work has shown that the gas-free vapor-pressure obtained experimentally agrees, in the case of all types of fuel, with the Reid vapor-pressure if we multiply the latter by 1.1. Further, both the gas-free and corrected Reid vapor-pressures can be accurately correlated with the 10-per cent point for fuels not containing special quantities of propane. In the case of Naturaline the agreement is 0.2 lb. per sq. in.

T. B. RENDEL[11]:—I am glad to see that Mr. Bridgeman will try experiments with actual carbureters. He will, I feel sure, find considerable diversity in his predicted results due to the fact that some carbureters can handle much more vapor than others and that we also have to recognize that in certain cases a carbureter may be set rich and therefore a slight decrease in the flow through the jet will not really affect the engine performance, which is our only gage of practical gas value. Supposing a slight interruption in flow occurs and the carbureter is set rich, we will have a little better fuel-economy.

I hope that before accepting his results we will see what the results on his practical experiments will be. My own experience is that the 10-per cent point for vapor pressure or any other test that we might like to make in that connection will give very variable results, unless we take our carbureter supply and the carbureter characteristics and design and the other points in the fuel system into consideration.

G. FRENCH[12]:—Do I understand that all gasoline is subject to vapor lock in the present fuel-systems? If we put on a pusher instead of a puller, will that change eliminate the trouble?

MR. BRIDGEMAN:—A fuel system can be designed which will cause vapor lock with any gasoline. In general, whether we will get vapor lock with a particular gasoline will depend upon the temperature and design of the fuel line. Numerous little points in the fuel line, if they can be improved, will materially decrease the tendency to vapor lock. Pushing the gasoline instead of raising it by suction lift would remove one of the most serious causes of trouble. Vapor lock is simply the result of gasoline being heated to the point where it can vaporize at a particular temperature and altitude. In other words, it is inherent in no one fuel but it is common to all fuels. Whether we get into trouble or not depends upon the volatility of the gasoline and upon the particular design of the fuel system.

MR. FRENCH:—Can we design a fuel system that would not have a tendency to vapor lock?

MR. BRIDGEMAN:—I think any designer can produce a fuel system that would work satisfactorily on the majority of the present-day aviation fuels. However, no matter how well a fuel line is designed, the refiner can still produce a fuel that will cause trouble. I do not think that there are fuels on the market at present which would vapor lock under normal conditions in the best type of system we could design.

Vapor Lock in Airplanes

BURNHAM ADAMS[13]:—I have done considerable traveling in the last year and have seen only two cases where I thought the trouble was due to vapor lock. One of these airplanes had a fuel system that was one of the

[10] See *National Petroleum News*, Feb. 5, 1930, p. 77.

[11] Jun. S.A.E.—Research engineer, Shell Petroleum Corp., St. Louis.

[12] A.S.A.E.—Assistant to manager of domestic lubricant sales, Sinclair Refining Co., New York City.

[13] Jun. S.A.E.—Engineer, Wright Aeronautical Corp., Paterson, N. J.

most wondrous things I ever gazed upon. Three of us with a diagram in hand spent three hours in tracing it out, but I cannot explain just how it works, although I have a general idea. I think that this fuel system was undoubtedly responsible for the total loss of this airplane a very few hours later. The other system I saw was where the manufacturer had endeavored to supply one of our 300-hp. motors through a ¼-in. gas line, and while the damage was not so severe in this case, he readily understood our words of wisdom enlightening him on the mistake he made. I think manufacturers on the whole appreciate the now-prevalent idea as to what causes vapor lock and are making every effort to avoid high spots in the fuel line and make the system as simple as possible.

Most of the present carbureters have the float-chamber back-suction mixture-control, in which case any pressure occurring in the float chamber will be regulated by the vent to the choke. I doubt if we will experience very much vapor-lock trouble on carbureters.

QUESTION:—Would not a closer understanding between the engine and the airplane designers in hooking up the engine be of great benefit in eliminating vapor lock?

MR. BRIDGEMAN:—I think so, for the problem of vapor lock is not limited to any one industry or designer. The engine designer cannot blame the petroleum industry solely for the vapor-lock troubles. Very frequently the fault lies in the other direction, but even that is not always true. It is a matter of cooperation between the two industries. Unquestionably we should get the best type of fuel for use in an optimum type or types of fuel system. If we do that and the fuel system is poorly designed, we are very apt to get into trouble due to vapor lock. With regard to the fuel, there are two things to consider and we have to strike a mean with some sacrifice in easy starting to avoid trouble from vapor lock, but a little better design of fuel systems would permit the use of more volatile fuels and hence better starting in winter. I might add that vapor lock is not necessarily confined to complete engine-stoppage, and from the standpoint of a large number of commercial operators who do not want to fly at very great altitudes, it may not be a very serious matter. In the case of those organizations, such as the Army, for instance, who do want to fly at great altitudes, a partial interruption of flow is a very serious matter and extremely minor changes in the fuel system may make an enormous difference. The Army is making some flight tests and temperature measurements during flight in different types of aircraft for us. In one of the tri-motored airplanes they recently found that one of the engines cut out at 10,000 to 12,000 ft. and did so consistently. Investigation showed that the fuel line leading to that engine had a bend, with a radius of about 4 to 6 in., where it went up over some part of the frame of the ship. By running the line straight across with a gradual slope that engine would operate satisfactorily at altitudes up to 16,000 or 18,000 ft. This was an extremely minor change in the fuel line, yet it remedied that difficulty. In many cases an extremely minor change in the fuel lines or systems has enabled the pilots to raise the ceiling of the airplane by several thousand feet.

DR. R. HASKELL[14]:—Has Dr. Bridgeman ever tried any experiments on selected condensation? I think aviation gasolines are the best stabilized of any gasolines we have, as we get at least 98 and at times 99 per cent recovery, which means that certain quantities not only of propane but also of other gases, which are really gases and not liquids, are dissolved in the gasoline, so that when distilled, those are driven off and do not go back. Gasoline with an initial boiling-point or condensing point of 100 deg. fahr., has to be heated to about 140 deg. to get anything to come off that will condense. The dissolved gas, that comes off at first, cannot be put back into the gasoline. We can shake it but it simply will not go back. Then these lighter condensable fractions commence to come off. In any experiments that are made we could let at least 10 per cent go by at the different temperatures before they condense and then see if that is any criterion. That would include the fixed gases, the propane and probably some other things. Would not that be a better criterion than taking the temperature of the vapor found on condensation?

MR. BRIDGEMAN:—In this preliminary work we started out to free the gasoline from the dissolved gases and then to take the vapor-pressure measurements. In addition to that we studied the effects of different quantities of dissolved gases on the vapor pressure. We found that the volume of dissolved gases was very small, air being the constituent dissolved in greatest measure, and 100 volumes of gasoline would hold about 15 volumes of air. Under normal conditions that would not seriously interfere with the gasoline flow if the line was so constructed that those bubbles of air would be eliminated in the carbureter or would pass up the line.

DR. HASKELL:—That is the only point I wanted to bring out to those who are not in the oil industry.

Vapor Lock and Airplane Crashes

A MEMBER:—I have read in the newspapers about the number of crashes, but have never been able to find out what the reasons were. Vapor lock might have been the cause for some of those crashes, but I have never heard it mentioned.

Does the Bureau of Standards feel that the temperature of 122 deg. fahr. for the 5-per cent point given in the present specifications for aviation gasoline is too high or should it be maintained?

MR. BRIDGEMAN:—I think that the 5-per cent point temperature is at present too high. How much that can be lowered depends entirely upon the temperatures which exist in typical aircraft under conditions of operation. That requires a large amount of data and takes much time to get. At present both the Army and Navy are cooperating with us in obtaining these data. When they have been obtained and analyzed, we can tell how much the specification limit can be lowered. I think unquestionably that the present limit does exclude gasolines which have been shown to give perfectly satisfactory performance in a large variety of airplanes.

To decide definitely whether crashes result from vapor lock is difficult. The Army, I believe, has a number of instances on record where airplanes have been forced down by vapor lock but did not necessarily end in a crash. As far as I can recall only one statement regarding an airplane having crashed with vapor lock has been published. I believe that Hinchcliffe when flying over the North Sea is supposed to have crashed due to vapor lock but that is the only press statement I have seen in regard to that.

[14] Industrial engineer, Texas Co., New York City.

A MEMBER:—The location of the gasoline strainer on most aircraft is a very important thing in regard to vapor lock. In our engines we always like to get a low strainer-installation as near the carbureter as we can so that we have the minimum of hose connection between the strainer and the carbureter. Thus we get the strainer very close to the source of heat, and the gasoline strainer represents a considerable volume of gas moving at a relatively low velocity and absorbing a considerable quantity of heat. What I get from those charts leads me to believe that for gasolines under those conditions, after leaving the gasoline strainer, to vapor lock would be comparatively easy; much more so than if the gas strainer were put in a cooler place so that the gasoline being delivered to the carbureter was at a relatively lower temperature. I think the location of the gasoline strainer in most of our airplanes should be given careful consideration. When it is up high and ahead of the fire-wall, it is bound to get pretty hot, and when we get down lower we run into the problem of trapping. The gasoline then has to rise a considerable distance to the carbureter.

MR. BRIDGEMAN:—In the data that we have obtained so far, the temperature of the gasoline in the carbureter float-bowl is higher than at other points in the system.

A MEMBER:—Eliminating the gas strainer, if that were possible, would still reduce the temperature on account of the fact that a considerable volume of gasoline is in the strainer. We have to use a much larger strainer than on the automobile engine on account of the vast quantity of fuel that must be delivered in a given time. We must also have a greater reserve of straining qualities, at least 100 per cent more than engine demands.

MR. BRIDGEMAN:—I might sum up about five things in connection with fuel lines that have grown out of this discussion. All unnecessary bends and connections should be eliminated. The size of the tubing should be twice what we think is necessary for the quantity of gasoline being delivered. All fuel lines and everything connected with them should be kept as cool as possible. The tank, which in many airplanes is well insulated thermally, should not be so insulated, because an airplane taking off from the ground with the gasoline at 100 deg. fahr. may reach 18,000 ft. with the gasoline then at 65 to 70 deg. fahr. even though the air temperature is below zero. The gasoline in the tank should be allowed to take up the temperature of the atmosphere more so than it does in a number of installations. Lastly, the fuel pump should be placed as low as possible; preferably, if a flexible drive is used, the fuel pump should be located on the same level or maybe a little lower than the gasoline in the tank. If the gasoline is fed under pressure, we have a decided advantage over a gravity-feed system. Consideration of some of these things that have grown out of our work might do much to reduce the tendency to vapor lock in airplane fuel-systems and it certainly would do considerable to increase the ceiling of the airplane, if that is desirable.

Gasoline Requirements of Commercial Aircraft Engines

(*Concluded from p.* 217)

other cases with 5 or 6 mg. the gasoline is frowned upon. The Army specifies 3 mg. but takes more, because it does not know; none of us knows. Mr. Cummings says the copper-dish test has not yet worked out. This larger committee is also to study methods of test. There is another point about the volatility. I do not think the engine manufacturers understand that they can have increased volatility when they make their systems so they can take it. They are all talking about 100-deg. fahr. initial boiling-point and things like that. There is no reason why they cannot have 80 and 70 and 60 deg. fahr. when they will make their fuel systems to take these more volatile fuels. High volatility does not cost any more than low volatility. I do not like the idea of the engine man saying that the oil company has not cooperated. The oil companies have cooperated. The engine manufacturer sets a limit. He does not design his engine so that it will take volatile fuel. That comes up in the vapor-locking question that Dr. Bridgeman[17] is to talk about.

Another point was brought up by Mr. Adams about antiknock in engines. We all know that the engine design can be changed to increase the compression ratio.

It is the engine manufacturers' responsibility to help this thing along; the oil companies will certainly aid. Why should not the engine manufacturers spend a little more money and bring out an engine that will burn ordinary fuel?

MR. DENHAM:—We ought to hear from the transport operators, too. This morning we heard the engine manufacturers placing the responsibility for fuel selection with the operators. Now we have the oil companies wanting to know what the engine manufacturers want in the way of fuel. We have not yet completed the cycle.

CHAIRMAN PECK:—We certainly should like to hear from some of these transport operators.

I think I voice the sentiment of the petroleum industry when I state this belief: In some quarters vast sums of money, we might say millions, are being spent annually in the petroleum industry in the endeavor to produce fuels of higher useful compression-ratio, and perhaps some of you gentlemen do not realize how hard we are working. We make fuels for 5.8:1 compression that are efficient and function beautifully. We walk out of the plant patting ourselves on the back and saying, "At last we have arrived." We go to another plant or to another manufacturer with the same fuel, and it has an H. U. C. R. of 5.4 perhaps.

That is worth thinking over. We want the suggestions and criticisms of the automotive industry. At the same time we should like to have it recognized that we are doing our part and all we want is to have the industry give a thought out of its wealth of experience that will enable us to go out with something definite that we can use so that we can make a better fuel and so that the engine manufacturers can make a better engine for a better airplane for better transportation. That is the problem we are all facing.

[17] See S.A.E. JOURNAL, July, 1930, p. 93.

Airplane Production Problems

St. Louis Aeronautic Meeting Paper

By William B. Robertson[1]

Routine and technique both present problems in the manufacture of airplanes. Routine, according to the author, includes the creation of a design, its acceptance by the sales department and other officials, the engineering release, scheduling by the production department, requisition and purchasing of parts, fabrication, inspection, testing and releasing to the sales department. Points stressed by the author with reference to technique, include the control of critical factors that require executive decision, the necessity for limiting overhead cost to a figure of not over 100 per cent of direct labor, and the segregation of service and repair work from the fabricating department.

The discussion was concerned with an explanation of the low figure for profit as given by the author, this figure being believed to have been based on the retail sales price of the completed airplane.

AIRPLANE production problems may be divided into two main phases, designated as production routine and production technique. Production routine includes the sequence of production in an organized factory, starting with the original conception of the design; its impression upon the officials who are competent to authorize sales estimates; its release from the engineering department; the steps in securing the raw material and passing it through the stock-room, the inspection department and factory management; and the assembly and delivery of the finished airplane into the hands of the sales or distributing organization. Production technique includes the control of the critical cost factors in building successful airplanes at a cost that is enough less than the manufacturer's selling price of the product to assure the successful operation of the company.

In discussing the production routine, I will assume that we are considering the manufacture of an airplane the experimental types of which have been built, which has been approved by the sales department and other executives and on which it has been agreed that production may proceed.

The first step is to determine the size of the initial production run. This run must not be so small that the advantage of tooling will be lost, neither must it be so large that the shop will be tempted to overtool, only to find that later changes and modifications necessitate scrapping the tools and replacing them later, with a duplication of expense. The policy must also be decided as to whether to build and stock or to build on customers' orders only. Needless to say, it is highly desirable to build for stock, wherever possible, but to keep the inventory of finished airplanes in stock at the absolute minimum. In the case of building on customers' order only, the question of how many to build is easily settled by the number of customers who have signed on the dotted line.

[1] M.S.A.E.—President, chairman of the board of directors, Curtiss-Robertson Airplane Mfg. Co., Robertson, Mo.

Initial Order Should Be for 25 Airplanes

In case of building for stock, which will some day be the predominating condition, a good first-production trial run is 25 machines. This gives the tool and production departments a real chance to give serious consideration to tooling the major and expensive items. It also gives the engineers sufficient opportunity to change details at a later date, which is always found desirable, and gives the shop departments the opportunity to incorporate labor-saving suggestions in the next run, with consequent economies in the manufacturing costs.

This choice of 25 machines for an initial run may be objected to by certain production experts on the grounds that good production control consists of monthly reorders, while the first production run of 25 machines on a new model cannot usually be completed within that time after the first plane has been delivered. However, strict adherence to the theory will, in this case, so limit the number of machines in the first run as to lose many of the benefits of the tooling, if the tooling is based on the savings effected only in the first one or two runs; and it may lead to serious loss in scrapping tools in case the tool designer and shop department gamble too far ahead on certain items only to find that they have been radically changed in subsequent production runs.

The quantity of the first production run having been thus determined, the engineering department is notified by competent authority to release the designated number of machines. This release, accompanied by the necessary blueprints and specifications, is then taken by the production-order department and the requisitions for the raw and purchased material are written in this department, delivered to the stock-room for filling —from minimum stock, if possible. If this is not possible, the requisitions are forwarded by the stores manager to the purchasing agent as authority to purchase. The purchasing agent returns the requisition to the

stock-room with a copy of his order as an indication to the stores manager of what material may be expected to arrive to fill the requisition or to replenish the minimum in the stock-room.

While this has been going on, the production-order department has also issued to the various shops the blueprints, together with the necessary shop-traveler or route tags, on which the production-order department has specified the routing. These instructions include the inspections necessary, the job order numbers to which work should be charged, and the requisition numbers on which the material for the job has been ordered and upon which it can be procured from the stock-room upon application. The shops are now ready to go to work.

The Shortage Dispatcher Is a Key Man

The department foremen assign the work among their men, procuring the necessary material from the stock-room and placing upon an urgent-shortage list any material not yet received or otherwise unobtainable. This shortage list is handled by a shortage dispatcher, whose sole duty it is to expedite such material from the vendor into the stock-room by prodding the purchasing agent, the vendor, the carriers, the inspectors who handle the material coming into the factory in bond, or by any other means necessary. This man is one of the vital links in a production factory; on the successful prosecution of his function may depend the making or the breaking of a company.

Let us now assume that the shops have fabricated the material; that it has been checked up by the routine dispatchers; has passed through the necessary inspection and, in the case of metals, the necessary processes such as heat-treatment, plating, internal oiling, and painting; and it is now ready for delivery to the assembly stock-room, to which it is delivered by the dispatcher.

Material from the assembly stock-room can be had by one of two systems, either with or without requisition. Undoubtedly the system with requisition is the more correct system; but, in view of the necessary flexibility of airplane production and the extreme necessity for low overhead in order to survive in the industry today, I believe the system of obtaining material from the assembly stock-room without a requisition is the more effective. There must, of course, be a number of safeguards in case the latter system is used—such as issuance of expensive items, like instruments, only to known personnel, and then in small quantities; but this system, which is in effect in some of the larger and more successful companies today, is in general the better for the purpose when properly safeguarded.

Having obtained the material from the assembly stock-room, the airplane is built and goes through the inspection department into the hands of the test pilot and from him to the sales distributing organization. This, in brief, is a rough outline of the general system of airplane production routine.

Scheduling the Year's Output

In the foregoing, we have discussed the phases of the first production order. Presumably, before this order is complete, the sales organization will be willing to release to the factory information on the anticipated requirements for the season. Let us assume that these requirements are released, and that the sales department says that it hopes to dispose of 500 machines of this type during the year. This figure, while larger than the number that the average factory will build, is not at all out of line. During 1929 there were three factories in this Country each of which built more than 500 airplanes.

Presumably the sales department will schedule these machines. To take up a definite example, let us assume that these machines will be desired on the following basis, by months: January, 25; February, 25; March, 50; April, 75; May, 75; June, 75; July, 50; August, 50; September, 40; October, 25; November, 5, and December, 5.

Let us assume also the ideal condition that the sales department is ready to estimate this production and that the first run of 25, which constitutes the 25 machines to be delivered in January, is released on October 1. This allows only 60 days for the production of the first machine and 90 days to complete the run, which is a short time, but at this time of the year all vendors are very anxious for business and can usually make very favorable deliveries. The same would be true of the February production, which should be released on Nov. 1. The March production should be released on Dec. 1 and the April production on Jan. 1. After the ball is once rolling, these date ranges of 60 to 90 days are quite possible.

The engineering department must have been competent, in the first place, to keep the vital errors at the minimum, and it must cooperate to keep the changes at the minimum. As far as possible, all changes must be made at the end of one production run and prior to the start of the next run. It even is more economical, in many cases, to continue to fabricate and install a part on which a replacement will later have to be made than it is to stop the production completely until the replacement part is available. Of course, no sacrifice is too much when life and death are concerned; but for the minor items, which constitute the majority of engineering changes, this is not the case.

All of the foregoing discussion has been on the supposition that a factory builds only one model, which is not true in any of the larger production factories. When more than one model is concerned, I believe none of the above factors are changed excepting that the unit runs of 25 machines on any one model may string out over a period of two to three months. The book authorities state that this is not a good production policy, but I believe that it is the best compromise possible under present conditions.

Production Technique

Labor, material and overhead are the three main items of airplane cost. Another classification used in the airplane business includes royalties, designers' percentages or development charges, which are not included with production overhead. The overhead is figured on the basis of the direct labor, and its percentage will be relatively high in localities where labor is cheap and low where labor is costly. The most successful airplane factories so far have had an overhead of not more than 100 per cent, including general, administrative and sales charges.

Such figures as are available seem to indicate that most airplane factories break even when their overhead is 100 per cent, and that the amount by which they can reduce the overhead below 100 per cent approximately

TABLE 1—DISTRIBUTION OF THE COST OF AN AIRPLANE

	Per Cent
Sales discount	25
Engine, cost to manufacture	30
Material other than the engine, cost to manufacture	20
Royalties, development costs and the like	5
Labor	8
Overhead	7
Profit	5

represents the net earnings in the year. Conditions make many variations; but it is a general rule that the balance sheet will have red-ink figures at the end of the year unless the overhead can be prevented from exceeding 100 per cent.

The ratio between labor and material also is an important feature in determining the ratio of labor to overhead. This labor-material ratio varies all the way from 5:1 in experimental military airplanes to 1:4 for small commercial airplanes that are in regular production. The cost of the engine is not included in these figures, but that of all other items is taken into account.

The list price of airplanes in ordinary commercial production in the United States today, built with new engines, would be allocated approximately as shown in Table 1. From this it appears that the typical commercial manufacturer today has only 20 per cent of the list price of his machine available for labor, overhead and profit. It is not difficult to see why the profit is almost sure to disappear when the overhead is much more than 100 per cent of the labor cost.

Segregate Service Work and Parts

Another important point in production technique is that it is absolutely necessary to see that repair work, service work and the sale of spare parts should not, under any condition, slow up a production schedule that has once been established and on which the costs are close. The best way of providing for emergency orders is to release spare parts specifically for storage against such a demand. The inventory required for this need not be large, provided that the variations in models are kept under reasonably good control. The usual spare-parts sale is an emergency sale resulting from some slight mishap which the customer is anxious to repair and forget.

Most customers will not object to paying a liberal price for their spare parts provided they can get immediate service; but they will object, seriously and with great justice, to being kept on the ground in some out-of-the-way town while the parts which they have been unfortunate enough to break are being fabricated at the factory. The release of repair parts should be controlled by the engineering department and the inventory should be allocated by the service manager, within a maximum which he is allowed by the management.

Major crash repairs on airplanes that are shipped to the factory are one of the worst obstacles to satisfactory production. If possible, a separate department should be organized to handle such work. This department should contain a few versatile mechanics and should draw upon the factory for regular work only in the emergencies.

THE DISCUSSION

CHAIRMAN E. P. WARNER[2]:—The figure of 8 per cent assigned to direct labor in the allocation of costs is extraordinarily low. With labor at 8 per cent, overhead at 7 per cent and profit at 5 per cent, there seems to be little hope for material economy from large-quantity production. I do not recall having ever seen any other business that was established or operated during its first stages on the assumption that the profit would be only 5 per cent of sales.

Since reading the preprint of this paper, I have consulted Moody's Manual and other authorities. I find that most of the automobile companies have made profits of 10 per cent on sales during the last two years and some have made about 20 per cent. Even the mail-order houses, which have a very large turnover, have made profits of from 7 to 10 per cent on gross business during the last two years. If the airplane business starts on the basis of a 5 per cent return, it should be classed as a philanthropy.

W. F. WISE[3]:—As a general rule, I believe that the ratio of profits is about 1:7. The 10-per cent figure in the automobile industry is based on an enormous turnover. Were the turnover small, I believe that the percentage would be higher. The first issue of the *Magazine of Wall Street* in January of each year gives the profit per car for the various automobile companies. Even telephone companies and other public utilities are asking for an 8-per cent return on capital, not on sales, and they are actually receiving it.

I believe that the explanation of the low percentage quoted by Mr. Robertson is that the percentages quoted are on the retail sales price, including the engine. I believe that the profit, to be comparable, should be based only on the direct labor, which he has given as 8 per cent; the overhead as 7 per cent; and the material, given as 20 per cent, making a total of 35 per cent of the retail sales price. I do not believe that the profit should be figured on the engine or on the sales discount. On this basis, the profit would be approximately 14 per cent of the cost of the airplane.

[2] M.S.A.E.—Editor of *Aviation*, New York City

[3] Sales manager, Ex-Cell-O Aircraft & Tool Corp., Detroit.

Automotive Research

Road Tests of Air-Cleaner Restriction

By A. H. HOFFMAN[1]

The Agricultural Engineering Division of the University of California has made three series of air-cleaner tests, one in 1922[2], one in 1924[3] and a third covering radiator fan-type cleaners in 1926[4]. The tests described in the present paper were conducted to compare the newer type of air-cleaner having a very low restriction with the type that has been on the market for a longer time.

Both dry and oil-type cleaners were tested on a nearly straight and level road at times of calm or very low wind-velocity, and the vacuum effects measured at car speeds from 20 to 60 m.p.h. The results are given in both graphical and tabular form.

MARKED progress in the development of air-cleaners having very low restriction has been made in the last two years. In an attempt to compare the restriction of the newer cleaners with that of some of those longer on the market, two series of road tests were made by the Agricultural Experiment Station at Davis, Calif., at speeds ranging from 20 to 60 m.p.h., one using a Buick 28-20 coach and the other using a Model A Ford coach. Ten dry-type cleaners (Fig. 1) and 10 oil-type (Fig. 2) were tried out on the Buick. Five cleaners, one dry-type and four oil-type (Fig. 3), all late models designed for the Model A Ford, were tried out on the car of that make. Data on the various cleaners are presented in Table 1.

The method employed when using the Buick was as follows: The cleaners in succession were attached to the carbureter by a short piece of tubing bearing a piezometer ring of conventional design connected to an ordinary open U-tube manometer. The manometer was carried in the car, connection being made by rubber tubes. The cleaner in each case was under the closed hood of the machine. In the case of some of the larger cleaners the use of longer tubular connections well taped and to make correction for their restriction was found necessary. As indicated in Table 2, the restriction of these tubes was considerable at the higher speeds. A nearly straight and level paved road was chosen and the tests run at times of calm or very low wind-velocity. Readings for each cleaner were taken going and returning and averaged to eliminate slight wind and grade effects. One adjustment of carbureter was maintained throughout.

In the tests of cleaners designed for the Model A Ford the method was in general the same as in those on the Buick, except that the cleaners were all attached directly to the carbureter inlet, as intended by the manufacturers of the cleaners, the piezometer connection being attached to the carbureter itself between the outer

[1] M.S.A.E.—Associate agricultural engineer, experiment station, University of California, Davis, Calif.

[2] See *Agricultural Engineering*, June, 1923, p. 89, and July, 1923, p. 109; also *Transactions of the American Society of Agricultural Engineers*, vol. 16, p. 50; and University of California Agricultural Experiment Station Bulletin No. 362.

[3] See THE JOURNAL, August, 1924, p. 140; also February, 1925, p. 249, and March, 1925, p. 337.

[4] See THE JOURNAL, July, 1927, p. 82.

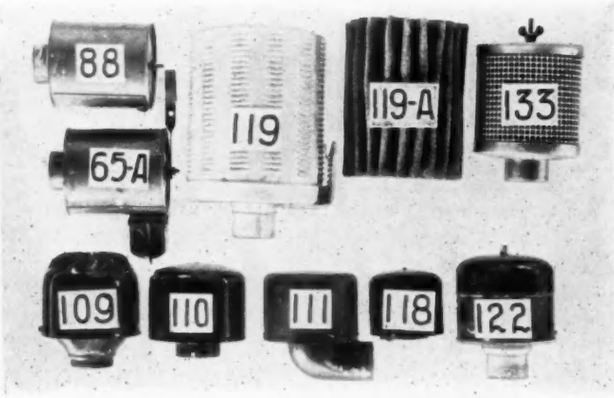


FIG. 1—DRY-TYPE CLEANERS TESTED ON BUICK

These Are: 65-*A*—AC Centrifugal Collector, 88—AC Centrifugal Ejector, 109—United Dry Centrifugal 4½-in. with Rotor, 110—United Centrifugal Ejector, 111—Handy Centrifugal Ejector, 118—United Centrifugal for Model A Ford, 119—Protectomotor C-4 Felt Filter, 119-*A*—Same as No. 119 but Used Four Years on Truck, 122—Tillotson Centrifugal Collector, and 133—Gordon Felt Filter

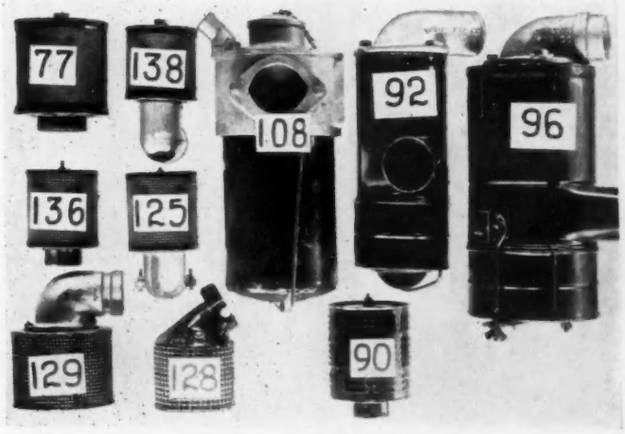


FIG. 2—OIL-TYPE CLEANERS TESTED ON BUICK

These Are: 77—Air-Maze 3 ST Wire-Screen Filter, 90—AC Copper-Ribbon Filter, 92—Vortox 850 Centrifugal and Self-Washing Kinked-Wire Filter, 96—Vortox 1000 Centrifugal and Self-Washing Kinked-Wire Filter, 108—Winslow Down-Flo Self-Washing Wire-Screen Filter, 125—National for Model A Ford Curled-Hair Filter, 128—Bowden for Model A Ford Pasteboard, 129—Bowden for Franklin Pasteboard, 136—National for Buick 28-20 and Curled-Hair Filter, and 138—Air-Maze for Model A Ford Wire-Screen Filter

TABLE 1—IDENTIFICATION OF AIR CLEANERS USED IN ROAD TESTS OF RESTRICTION

No.	Name and Model	Made By	Weight When Clean and Dry, Lb.	Inside Diam. of Outlet, In.	Height and Diam. of Body Proper, In.	Elbow in Outlet	Material	
							Body	Filter, if Any
65-A	AC (1926) Dry Centrifugal Collector	AC Spark Plug Co. Flint, Mich.	1.00	1 13/16	4 1/2 x 4	No	Sheet Steel	
77	Air-Maze 3 ST Oily-Screen Filter	Air-Maze Corp., Cleveland	1.30	2 3/8	4 5/8 x 4 5/8	No	Pressed Steel	Wire Screen
88	AC Dry Centrifugal Ejector	AC Spark Plug Co. Flint, Mich.	0.60	1 3/4	4 5/8 x 4	No	Pressed Steel	
90	AC Oily Copper-Strip Filter	AC Spark Plug Co., Flint, Mich.	0.75	1 13/16	4 1/2 x 4 1/8	No	Pressed Steel	
92	Vortox 850, Self-Washing Filter	Vortox Mfg. Co., Claremont, Calif.	4.90	2	11 x 4 1/2	Yes	Cast Aluminum Top Pressed Steel	Kinked Steel-Wire
96	Vortox 1000 Self-Washing Filter	Vortox Mfg. Co., Claremont, Calif.	7.70	2	14 1/2 x 6 1/2	Yes	Cast Aluminum Top Pressed Steel	Kinked Steel-Wire
108	Winslow Down-Flo Self-Washing Screen Filter	Michiana Products Corp., Michigan City, Ind.	10.65	2 1/4	12 x 6	No	Aluminum and Sheet Steel	Wire Screen
109	United Dry Centrifugal with Rotor	United Air Cleaner Corp., Chicago	1.40	1 5/16	4 x 4 1/2	No	Sheet Steel	
110	United, 4 1/2 In. Horizontal Dry Centrifugal, 4 Slots	United Air Cleaner Corp., Chicago	1.13	2 1/16	3 1/2 x 4 1/2	No	Sheet Steel	
111	Handy Dry Centrifugal, 1 Slot	Handy Cleaner Corp., Detroit	0.80	1 13/16	3 1/8 x 4 3/8	Yes	Sheet Steel	
118	United for Model A Ford, Dry Centrifugal, 4 Slots	United Air Cleaner Corp., Chicago	0.62	1 1/2	3 x 3 1/2	Yes	Sheet Steel	
119	Protectomotor, C-4, Dry-Felt Filter	Staynew Filter Corp., Rochester, N. Y.	2.50	1 7/8	7 1/2 x 6 1/4	No	Pressed Aluminum	Felt About 1/16 In. Thick
119-A[5]	Protectomotor C-4 Dry-Felt Filter							
122	Tillotson X-1 Dry Centrifugal Collector	Tillotson Mfg. Co., Toledo	1.13	1 7/8	4 x 4 1/2	No	Pressed Steel	
125	National for Model A Ford Oily Hair-Filter	National Air Filter Co., Dayton, Ohio	0.65	1 3/4	4 x 3 1/2	Yes	Pressed Steel and Hwd. Cloth	Loosely Packed Curled-Hair
128	Bowden for Model A Ford Oily Pasteboard	J. A. Bowden, Los Angeles, Calif.	0.75	1 7/8	4 x 4	Yes 45 Deg.	Hwd. Cloth and Sheet Steel	Pasteboard, 3/32 In.
129	Bowden for Franklin, Oily-Pasteboard Filter	J. A. Bowden, Los Angeles, Calif.	2.20	1 13/16	4 x 5	Yes	Aluminum Top, Hwd. Cloth	Pasteboard, 3/32 In.
133	Gordon 5-In. Dry-Felt Filter	Gordon Air Filter Co., Kansas City, Mo.	1.75	2	5 x 5	No	Sheet Steel and Hwd. Cloth	Felt 1/4 In. Thick
136	National for 1928 Buick, Oily-Hair Filter	National Air Filter Co., Dayton, O.	0.75	1 13/16	3 7/8 x 3 1/2	No	Pressed Steel and Hwd. Cloth	Curled Hair
138	Air-Maze, for Model A Ford, Oily-Screen Filter	Air-Maze Corp., Cleveland	1.25	1 5/8	6 1/2 x 3 1/2	Yes	Pressed Steel and Cast Aluminum	Wire Screen
143	Bowden for Model A Ford, Oily-Pasteboard Filter	J. A. Bowden, Los Angeles, Calif.	1.40	1 3/4	5 1/2 x 4	Yes	Sheet Steel, Hwd. Cloth and Cast Aluminum	Pasteboard, 3/64 In.
146	Bowden for Model A Ford Oily-Pasteboard Filter	J. A. Bowden, Los Angeles, Calif.	1.45	1 5/8	5 1/4 x 4	Yes	Pressed Steel, Hwd. Cloth and Cast Aluminum	Pasteboard, 3/64 In.

[5] This cleaner is like No. 119 but had been used on a truck for 4 years before it was given the road test for vacuum, the results of which are presented in Table 2 and Fig. 4.

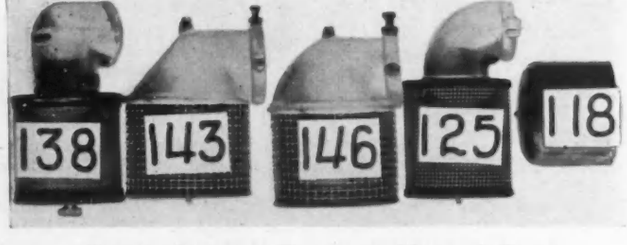


FIG. 3—CLEANERS TESTED ON MODEL A FORD

These Are: 118—United Dry Centrifugal without Rotor, 125—National Oily Curled-Hair Filter, 138—Air-Maze Oily-Screen Filter, and 143 and 146—Bowden Oily Pasteboard

end of the inlet and the choker valve, and that a test of maximum possible-speed was included. It should be pointed out that these were in no sense tests of the cars but of the cleaners only.

The results of the tests using the Buick are given in Table 2 and shown graphically for the dry-type cleaners in Fig. 4 and for the oil type in Fig. 5. As will be observed, the curves are drawn smoothly, ignoring slight discrepancies due to deviations from level in the road and changes in wind velocity. As would be expected, these discrepancies are more apparent at low car-speeds. It is believed that they were practically negligible at speeds above 40 m.p.h.

The results given in Table 2 and Figs. 4 and 5 indi-

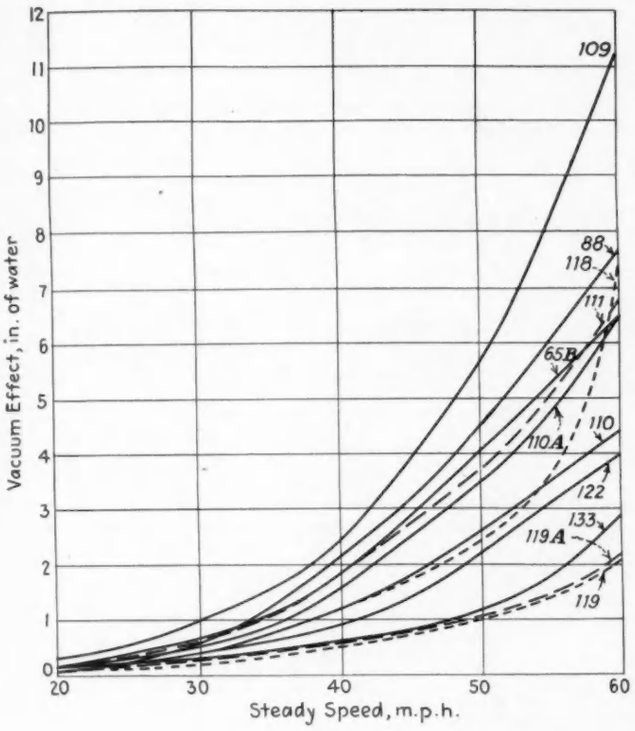


TABLE 3—ROAD TESTS OF AIR-CLEANER RESTRICTION AND EFFECT ON CAR SPEED[10]

No.	Cleaner (All are Models Designed for Model A Ford)	Vacuum, In. of Water, at M.P.H.							Highest Speed Obtainable, M.P.H.
		20	30	40	50	58	59	60	
118	United Dry Centrifugal Ejector 4 Slots	0.1	0.7	1.5	3.9	....	9.5	....	59
125	National Loose Oily-Hair	0.1	0.4	1.0	2.6	....	....	7.0	60
138	Air-Maze Oily Screen	0.1	0.5	1.0	2.6	....	....	7.5	60
143	Bowden Oily Pasteboard	0.1	0.5	1.1	3.1	8.1	....	....	58
146	Bowden Oily Pasteboard	0.1	0.4	1.2	3.0	8.4	....	....	58
	None	....	....	....	....	....	....	....	60.5

[10] The tests were made on a Model A Ford coach that had run 7000 miles and were over a nearly level paved highway when the air was calm. During the tests the car carried a driver and two observers. The barometer reading was 30.000 in., the air temperature ranged from 50 to 56 deg. fahr. and the humidity was 55 per cent. The cleaners were free from dust and were designed for Model A Fords, descriptive data being given in Table 1 and illustrations in Fig. 3. The carbureter adjustment was left unchanged throughout the tests.

FIG. 4—ROAD TESTS OF VACUUM

Dry-Type Cleaners either Clean or Freshly Serviced, except 119-*A*, Were Applied in Succession to Carbureter. Car Was Run in Calm Air on Level Road and Vacuum Effect Was Measured at Speeds Ranging from 20 to 60 M.P.H. The Cleaners Were: 65-*B*—AC Collector, 88—AC Ejector, 109—United with Rotor, 110—United with Inlet Facing Fan, 110-*A*—Same with Inlet Facing Away from Fan, 111—Handy, 118—United for Model A Ford, 119—Protectometor, 119-*A*—Same but Used Four Years on Truck, 122—Tillotson, 133—Gordon

TABLE 2—ROAD TESTS OF AIR-CLEANER RESTRICTION MADE ON A BUICK 28-20 COACH

Cleaner		Outlet Elbow	Vacuum, In. of Water, at M.P.H.					Remarks
No.[7]	Make and Type		20	30	40	50	60	
65-*A*	AC Dry Centrifugal Collector off 1926 Buick	No	0.2	1.0	1.8	4.1	6.5	Old Style with Cup
77	Air-Maze Oily Screen	No	0.1	0.1	0.4	0.7	2.0	
88	AC Dry Centrifugal Ejector off Buick 28–20	No	0.2	0.4	2.5	4.4	7.7	
90	AC Oily Copper-Ribbon	No	0.1	0.3	1.2	2.0	2.9	
92	Vortox 850 Self-Washing Oily-Wire	Yes	0.5	1.7	3.2	6.7	12.8	Some Oil Went Over at 60 M.P.H.
96	Vortox 1000 Self-Washing Oily-Wire	Yes	0.3	0.9	2.3	4.9	8.2	
108	Winslow Down-Flo Self-Washing Oily Screen	No	0.1	0.3	1.3	2.5	3.5	Trace of Oil Went Over at 60z M.P.H.
109	United Dy Centrifugal Ejector with Rotor	No	0.3	1.0	2.4	5.6	11.3	
110	United Dry Centrifugal Ejector without Rotor	No	0.1	0.4	1.4	2.6	4.4	Inlet Facing Air-Stream
110-*A*	United Dry Centrifugal Ejector without Rotor	No	0.2	0.5	1.6	3.5	6.5	Inlet Facing Down Air-Stream
111	Handy Dry Centrifugal Ejector	Yes	0.2	0.6	1.8	3.7	6.8	
118	United Dry Centrifugal Ejector without Rotor for Mod. A Ford	No	0.2	0.4	1.2	2.4	7.4	
119	Protectomotor C-4 Dry Felt	No	0.1	0.2	0.6	1.5	2.1	
119-*A*	Protectomotor C-4 Dry Felt	No	0.1	0.3	0.7	1.7	2.2	After 4 years' Use on Extension Division Truck
122	Tilloston X-1 Dry Centrifugal Collector	No	0.1	0.4	0.9	2.5	4.0	
125	National Loose Oily-Hair for Model A Ford	Yes	0.2	0.5	2.0	3.4	6.0	
128	Bowden Oily Pasteboard for Model A Ford	Yes 45 Deg.	0.2	0.6	2.3	3.5	6.0	
129	Bowden Oily Pasteboard for Franklin	Yes	0.1	0.3	1.2	2.5	4.4	
133	Gordon Dry Felt	No	0.1	0.3	0.6	1.2	2.9	
136	National Loose Oily-Hair for Buick 28–20	No	0.1	0.2	1.1	2.0	4.9	
138	Air-Maze Oily Screen for Model A Ford	Yes	0.1	0.4	1.0	2.0	4.7	
Tube *A*	Flexible Metal Hose, 2¼ In. Inside Diameter and 36 In. Long	....	0.2	0.4	1.0	2.2	3.3	Used as Connection to Carbureter[8]
Tube *B*	Steel, 1⅝ In. Inside Dia. and 24 In. Long with Brazed Elbows	Two 45 Deg.	0.1	0.3	0.8	1.7	3.0	Used as Connection to Carbureter[9]

[6] This test was made on a nearly level paved road when the air was nearly calm. The readings are the average of those taken for travel in opposite directions. All cleaners were free from dust and freshly serviced unless otherwise specified. The carbureter adjustment was left unchanged throughout the tests.
[7] See Table 1 for descriptive data and Figs. 1 and 2 for illustrations.
[8] For air-cleaners Nos. 92, 96, 108, 119 and 119-*A*.
[9] For one installation of air-cleaner No. 129.

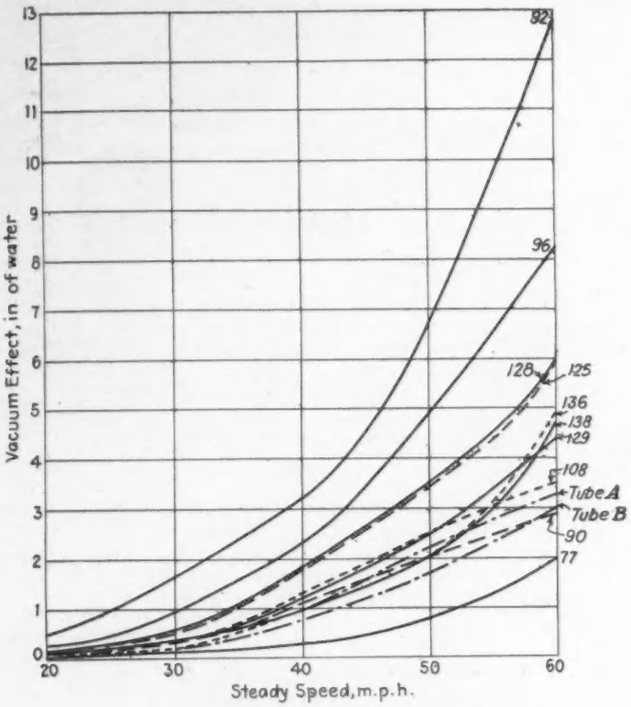


FIG. 5—ROAD TESTS OF VACUUM

Oily-Filter Type Cleaners Freshly Serviced Were Used in This Test, Conditions Being the Same as for the Dry-Type Tests. The Cleaners Tested Were: 77—Air-Maze, 90—AC Copper Ribbon, 92—Vortox 850, 96—Vortox 1000, 108—Winslow Down-Flo, 125—National for Model A Ford, 128—Bowden for Model A Ford, 129—Bowden, 136—National, and 138—Air-Maze for Model A Ford. Two Tubes Were Used; *A*, Metal Hose 2¼ In. Inside Diameter and 36 In. Long, and *B*, Seamless Steel 1⅝ In. Inside Diameter and 24 In. Long with Two 45-Deg. Elbows

cate that at and below 40 m.p.h. and with the cleaners freshly serviced none of the cleaners tested offers enough restriction to interfere seriously with the action of most types of carbureter; but that at higher speeds some of them, especially those that have centrifugal action, increase in restriction very rapidly. This is due in part to the fact that energy is required to set air into whirling motion. The kinetic energy in the whirling air would be obtained at the expense of a drop in pressure. The "air straighteners" placed in the outlet tubes of certain dry centrifugal air-cleaners serve to restore some of this lost pressure. The presence of an elbow in the outlet of a cleaner often markedly increases the restriction effect. This is exemplified in the tests of Air-Maze cleaners Nos. 77 and 138 of Table 2 and Fig. 5, the former having an elbow while the latter has not. Similarly, National No. 136, although the same size as 125 of the same make, shows less restriction because it is not handicapped by an elbow.

Whether the air inlet of a cleaner faces the blast from the radiator fan or down stream influences noticeably the net restriction. Thus cleaner No. 110, a United dry centrifugal without rotor, was tested in the two positions, one facing the air stream (110) and one facing down stream (110-*A*). Both these placings for dry centrifugals are found in practice. As will be noted in Table 2 and Fig. 4, the down-stream facing raised the vacuum effect by about 50 per cent. Nevertheless, this placing is preferable because the quantity of dust entering the cleaner when located in this position is much less and therefore the over-all efficiency is appreciably higher.

[11] See THE JOURNAL, March, 1927, p. 393.

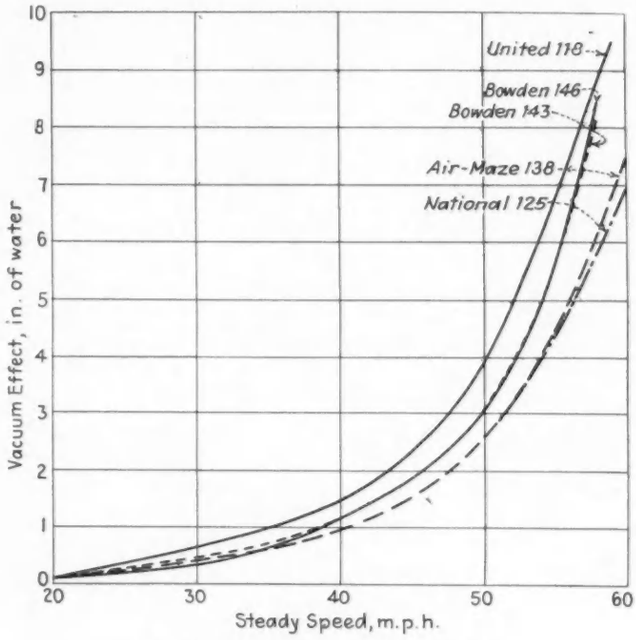


FIG. 6—ROAD TESTS OF VACUUM

All Cleaners Were Designed for Model A Ford and All Tests Were Run on the Same Half-Day on the Same Ford Coach, the Test Conditions Being the Same as for the Other Two Series of Tests. Carbureter Adjustment Was Not Touched during the Tests

In the tests using the Buick no attempt was made to find the maximum speed attainable. Using the Model A Ford we found, as shown by Table 3 and Fig. 6, that none of the cleaners tested had very much effect on the maximum speed. With no cleaner the maximum speed that could be maintained on the level was 60.5 m.p.h.; with cleaners Nos. 125 and 38, 60 m.p.h.; with No. 118, 59 m.p.h.; and with Nos. 143 and 146, 58 m.p.h. Cleaner No. 118 was placed, as designed, with its air inlet facing the fan blast. The slight supercharging effect thus produced evidently overcame to some extent the pressure drop produced because of the centrifugal action.

It should be borne in mind in a study of Tables 2 and 3 and Figs. 4, 5 and 6 that all the cleaners tested, except No. 119-*A*, were clean of dust when given the road test. Marked differences would have been shown if each cleaner had been given a similar test after 10,000 miles of ordinary use on an automobile. Some cleaners increase their restriction effect very rapidly as dust is taken in; others are affected but slightly or not at all. The characteristic behavior of the different types has been reported[11].

Standardization Progress

Antiknock Tests for Gasoline

Definite Basis for Reference Numbers Proposed for Approval as S.A.E. Recommended Practice

BOTH users and producers of gasoline fuel have been dissatisfied for some time with the unsatisfactory systems of fuel knock-test measurements, which have made it impossible to compare results obtained in one laboratory with those of another because of the diversity of apparatus, methods and reference standards used. During 1927 the problem was discussed at meetings of the Cooperative Fuel-Research Steering Committee, which is a joint committee of the American Petroleum Institute, the National Automobile Chamber of Commerce and the Society. In February, 1928, a subcommittee was appointed to develop apparatus for a method of rating knock and a standard reference fuel or a scale of fuels that would make knock-testing results comparable and that would be generally used as standard.

The members of the subcommittee were in agreement on the main details of the program and that the first step to be taken was to develop a suitable engine for knock-testing, so that all experimental work on methods of rating fuel for knock would be on a common basis. During the last year the laboratories represented by the committee[1] have concentrated on perfecting a test engine and studying the effects of engine variables on the knock ratings of a series of test fuels. Determinations are in progress on a group of methods[2] representative of those now commonly used, for the purpose of deciding upon a standard procedure. To make more rapid progress in the preliminary work and to have at least a temporary standard basis for testing, the Detonation Subcommittee, at a recent meeting, unanimously adopted the following resolution embodying its views:

Detonation Subcommittee Resolution

That the Detonation Subcommittee makes more definite its announced intention to refer knock-testing results back to heptane-octane by using a scale of octane numbers, the octane numbers to be the number of parts of octane by volume added to 10 parts by volume of normal heptane; and proposes to the Cooperative Fuel-Research Steering Committee that the companies represented on the Steering Committee be invited to follow the example of this Committee and to interpret their methods of rating fuels in the same language, recognizing that such ratings will be somewhat approximate for the present until standard methods of test have been generally agreed upon; and recommends to the S.A.E. Research Committee that this method of specifying antiknock fuels be approved as a standard practice.

Engine Builders Approve Proposal

The resolution of the Subcommittee was approved by the Research Committee and referred to the Standards Committee with the request that the latter proceed in the regular way toward adoption of the Detonation Subcommittee's resolution as S.A.E. Recommended Practice. The Council has been requested to assign the subject to the Lubricants Division of the Standards Committee, and it has been referred to the Division for action by letter ballot. The proposal was also sent to the principal engine builders and to automobile companies that make engines, with a request for their comments. Replies from a small majority of the Division members had been received by July 10, nearly all of them approving the proposal in general and in some instances suggesting slight modifications in wording. The proposal being considered by the Lubricants Division is

S.A.E. ANTIKNOCK NUMBERS

(Proposed for S.A.E. Recommended Practice)

The antiknock value of fuels shall be specified by referring detonation-test results back to heptane-octane, using a scale of octane numbers; these octane numbers to be the number of parts by volume of octane added to 10 parts by volume of normal heptane.

Note: This recommendation is made recognizing that such ratings will be somewhat approximate for the present and until standard methods of test have been generally agreed upon.

As soon as the Division has recorded a more complete vote on the proposal, it will be prepared in the form approved for submission to the Standards Committee and Society at the Annual Meeting next January for adoption as S.A.E. Recommended Practice. In the meantime the Detonation Subcommittee will proceed with its tests on the basis of the recommendation as approved by the Lubricants Division. Further information regarding the project can be obtained from the Research Department of the Society.

[1] See S.A.E. JOURNAL, February, 1929, p. 212.

[2] See S.A.E. JOURNAL, July, 1929, p. 80.

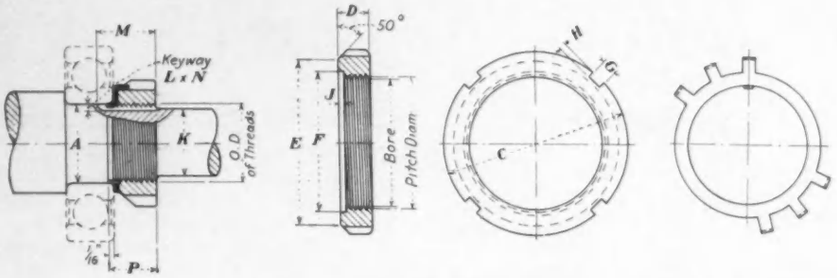
Ball-Bearing Lock-Nuts and Washers

Subdivision of Ball and Roller-Bearings Division Proposes an S.A.E. Standard

STANDARDIZATION of the boundary dimensions of several types of ball-bearing, especially those used by the automotive industries, was effected by the Standards Committee nearly 20 years ago. These standards, which have been refined by the Ball and Roller-Bearings Division of the Standards Committee a number of times since then, were the basis for the present American Standard for the single-row annular bearings in the light, medium and heavy series.

At a recent meeting of the Division it was indicated that the bearing locking-nuts and washers made by several of the bearing manufacturers are so nearly alike that the manufacturers, as well as the bearing users, would be greatly benefited by more complete standardization of these nuts and washers. Accordingly, a Subdivision consisting of H. N. Parsons, of the Strom Bearings Co., Chairman; H. E. Brunner, of the S.K.F. Industries; and E. R. Carter, Jr., of the Fafnir Bearing Co., was appointed to study the subject and submitted its report for consideration by the Ball and Roller-Bearings Division.

The principal dimensions in the report governing the design and interchangeability of the lock-nuts and washers is published in this issue of the S.A.E. JOURNAL for study by bearing manufacturers and users and to give them an opportunity to comment on it before the Division makes its final recommendation to the Standards Committee and the Society for adoption as a standard. Comments should be addressed to the Standards Department.



pleted its report after several years of careful investigation and study.

The report includes nomenclature for roller-chain parts and dimensions for standard and extra-heavy series of chains. The standard for these chains has been worked out mathematically in accordance with formulas given in the report and on the basis of the experience of the chain manufacturing industry. Valuable data governing the design of standard sprocket teeth and the cutters for them are also included. The new American Standard, which includes only the chains, sprockets and cutters, is supplemented by formulas and tables for working loads and horsepowers for the standard chains, as well as pitch diameters and outside diameters of sprockets and information relating to patterns for cast sprockets.

This report has been approved by the Society as a sponsor for the Sectional Committee and its project, and adoption of the standard will in effect supersede the S.A.E. Standard for Roller Chains, commencing on p. 103 of the 1930 edition of the S.A.E. HANDBOOK.

The report is being printed in pamphlet form 8½ x 11 in. in size. Copies can be obtained from the Society at nominal cost. Its official designation is A.S.A. No. B29a-1930.

BALL-BEARING LOCK-NUTS AND WASHERS

(Proposal for S.A.E. Standard)

Lock-Nut No.	Bearing Bore	Nut							Shaft				Threads, U. S. Standard Form				
		Bore *B*	O. D. *C*	Width *D*	O. D. of Face *E*	Relief Diam. *F*	Slot Width *G*	Relief Depth *J*	Ext. Shaft *K* Max.	Keyway Depth *L*	Keyway Width *N*	Thread Length *P*	Pitch	Shaft		Nut	
														Max.	Min.	Max.	Min.
0	0.3937	0.3541	3/4	7/32	1/2	13/32	1/8	1/32	5/16	1/16	1/8	5/16	32	0.3707	0.3677	0.3747	0.3717
1	0.4724	0.4321	7/8	5/16	23/32	1/2	1/8	1/32	13/32	1/16	1/8	13/32	32	0.4487	0.4457	0.4529	0.4497
2	0.5906	0.5491	1	5/16	53/64	5/8	1/8	1/32	1/2	5/64	1/8	1/2	32	0.5657	0.5627	0.5697	0.5667
3	0.6693	0.6271	1 1/8	11/32	29/32	23/32	1/8	3/64	9/16	5/64	1/8	7/16	32	0.6437	0.6407	0.6477	0.6447
4	0.7874	0.7441	1 3/8	3/8	27/32	27/32	3/16	1/16	23/32	5/64	3/16	15/32	32	0.7607	0.7577	0.7647	0.7617
5	0.9843	0.9321	1 9/16	13/32	1 9/32	1	3/16	1/16	7/8	3/32	3/16	17/32	32	0.9487	0.9457	0.9527	0.9497
6	1.1811	1.1073	1 3/4	13/32	1 1/2	1 7/32	3/16	1/16	1 1/16	3/32	3/16	17/32	18	1.1359	1.1329	1.1399	1.1369
7	1.3780	1.3103	2 1/16	7/16	1 13/16	1 7/16	3/16	1/16	1 1/4	3/32	3/16	9/16	18	1.3389	1.3359	1.3429	1.3399
8	1.5748	1.4973	2 1/4	7/16	2	1 5/8	1/4	1/16	1 15/32	3/32	5/16	9/16	18	1.5259	1.5229	1.5299	1.5269
9	1.7717	1.7013	2 17/32	7/16	2 9/32	1 13/16	1/4	1/16	1 11/16	3/32	5/16	9/16	18	1.7299	1.7269	1.7339	1.7309
10	1.9685	1.9033	2 11/16	1/2	2 7/16	2	1/4	1/16	1 7/8	3/32	5/16	5/8	18	1.9319	1.9289	1.9359	1.9329
11	2.1654	2.0913	2 31/32	1/2	2 21/32	2 3/16	1/4	3/32	2 1/16	3/32	5/16	5/8	18	2.1199	2.1169	2.1239	2.1209
12	2.3622	2.2943	3 5/32	17/32	2 27/32	2 13/32	1/4	3/32	2 1/4	3/32	5/16	21/32	18	2.3229	2.3199	2.3269	2.3239
13	2.5591	2.4823	3 5/8	9/16	3 1/16	2 19/32	1/4	3/32	2 7/16	3/32	5/16	11/16	18	2.5109	2.5079	2.5149	2.5119
14	2.7559	2.6853	3 5/8	9/16	3 5/16	2 13/16	1/4	3/32	2 5/8	3/32	5/16	11/16	18	2.7139	2.7109	2.7179	2.7149
15	2.9528	2.8300	3 7/8	19/32	3 9/16	3	3/8	3/32	2 25/32	1/8	5/16	23/32	11	2.8780	2.8740	2.8830	2.8790
16	3.1496	3.0340	4 5/32	19/32	3 27/32	3 3/16	3/8	3/32	3	1/8	3/8	23/32	11	3.0820	3.0780	3.0880	3.0840
17	3.3465	3.2370	4 13/32	5/8	4 1/32	3 3/8	3/8	3/32	3 3/16	1/8	3/8	3/4	11	3.2850	3.2810	3.2910	3.2870
18	3.5433	3.4240	4 21/32	11/16	4 9/32	3 19/32	3/8	3/32	3 3/8	1/8	3/8	27/32	11	3.4720	3.4680	3.4780	3.4740
19	3.7402	3.6270	4 25/32	23/32	4 9/16	3 25/32	3/8	3/32	3 9/16	1/8	3/8	7/8	11	3.6750	3.6710	3.6810	3.6770
20	3.9370	3.8150	5 3/16	3/4	4 13/16	4	3/8	3/32	3 25/32	1/8	3/8	29/32	11	3.8630	3.8590	3.8690	3.8650
21	4.1339	4.0180	5 7/16	3/4	5	4 7/32	1/2	3/32	3 15/16	1/8	3/8	29/32	11	4.0660	4.0620	4.0720	4.0680
22	4.3307	4.2210	5 23/32	25/32	5 9/32	4 13/32	1/2	1/8	4 3/16	5/32	3/8	31/32	11	4.2690	4.2650	4.2750	4.2710
23	4.5276																
24	4.7244	4.6120	6 1/8	13/16	5 11/16	4 13/16	1/2	1/8	4 9/16	5/32	3/8	1	11	4.6600	4.6560	4.6660	4.6620
25	4.9213																
26	5.1181	5.0020	6 3/4	29/32	6 3/16	5 7/32	5/8	1/8	4 15/16	5/32	1/2	1 3/32	11	5.0500	5.0460	5.0560	5.0520
27	5.3150																
28	5.5118	6.3930	7 3/32	15/16	6 17/32	5 19/32	5/8	1/8	5 5/16	5/32	1/2	1 1/8	11	5.4410	5.4370	5.4470	5.4430
29	5.7087																
30	5.9055	5.7840	7 11/16	1	7 1/16	6	5/8	1/8	5 23/32	3/16	1/2	1 7/32	11	5.8320	5.8280	5.8380	5.8340
31	6.1024																
32	6.2992	6.1900	8 1/16	1 1/32	7 7/16	6 3/8	5/8	1/8	6 1/8	3/16	1/2	1 1/4	11	6.2380	6.2330	6.2460	6.2410
33	6.4961																
34	6.6929	6.5800	8 21/32	1 1/16	8 1/32	6 25/32	5/8	1/8	6 1/2	3/16	1/2	1 9/32	11	6.6280	6.6230	6.6360	6.6310
35	6.8898																
36	7.0866	6.9710	9 1/16	1 1/8	8 3/8	7 5/32	3/4	1/8	6 29/32	3/16	1/2	1 11/32	11	7.0190	7.0140	7.0270	7.0220
37	7.2835																
38	7.4803	7.3620	9 15/32	1 1/8	8 25/32	7 9/16	3/4	1/8	7 5/16	3/16	1/2	1 11/32	11	7.4100	7.4050	7.4180	7.4130
39	7.6772																
40	7.8740	7.7520	9 27/32	1 3/16	9 5/32	8	3/4	1/8	7 11/16	3/16	1/2	1 11/32	11	7.8000	7.7950	7.8080	7.8030
41	8.0709																
42	8.2678																
43	8.4646																
44	8.6615																

Standard for Roller Chains

Report of Sectional Committee Finally Approved for Adoption as American Standard

WHAT was probably the first step toward the standardization of roller chains was a data sheet printed in the S.A.E. HANDBOOK in 1913 giving formulas for calculating the lengths of roller chains based on a series of chain pitches, roller diameters and widths, and sprocket-tooth outlines. These data were primarily for roller chains to be used in motor-vehicle drives.

About four years later, however, further standardization of roller chains was taken up jointly by the Society and the American Society of Mechanical Engineers and continued until 1920. In 1921 specifications were also adopted for roller-chain sprockets and sprocket cutters.

In 1924 the Sectional Committee on Transmission Chains, Sprockets and Cutters was organized under the procedure of the American Engineering Standards Committee, now the American Standards Association, and sponsored by the Society, the American Society of Mechanical Engineers and the American Gear Manufacturers Association. This Committee has com-

Personal Notes of the Members

Rippingille Returns from Germany

Having acted as executive engineer for the General Motors G.m. b.H., at Berlin, Germany, since 1928, E. V. Rippingille recently returned to his home in Michigan to take a two months' rest. His mail address will be in care of the Detroit office of the General Motors Export Co. During the last 2½ years he has been located at the Berlin headquarters of the Export company and has been engaged in organizing an engineering department having to do with the establishment of standards for bodies for General Motors Corp. trucks and motorcoaches throughout the world. He has also been engaged in extending the parts export service and the setting up of a purchasing center in Berlin for materials and parts for the corporation's plants and depots throughout the world. This involved familiarizing himself with the sources of supply for all such materials. He has now come home to take a well-earned vacation.

Born at Perth, Australia, in 1885, Mr. Rippingille attended the high school in that city and later the Technical College at London, England, receiving a first-class diploma in general mechanics and bridge construction from the South Kensington Technological Institute. Thereafter, he was engaged in engine drafting, testing and experimental work with the Societe Lorraine de Dietrich et Cie., at Luneville and Paris, France. Subsequently he was connected for a year with the de Dietrich Import Corp., of New York City, in the management of that firm's general repair business. His next position was with the Bristol Engineering Corp., of Bristol, Conn., where his work dealt mainly with engine experiments and road testing.

Late in 1910, Mr. Rippingille joined the Hudson Motor Car Co., with which he remained for 11 years, occupying various engineering and executive positions. In 1922 he became president and treasurer of the Watson Stabilator Co., of Detroit, and three years later accepted the post of sales manager for the Dayton Engineering Laboratories Co., of Dayton, Ohio. After the organization of the Delco-Remy Corp. he was a sales engineer for the corporation until, in 1928, the General Motors Export chose him for regional engineer, with headquarters at Berlin.

A Member of the Society since 1910, Mr. Rippingille has served on many committees and also been prominent in Sections work. He was a member of the Spring Division of the Standards Committee during 1922 and 1923; Chairman of the Frames Division in 1922; a member of the Sections Committee in 1921, and treasurer of the Detroit Section in 1925 and 1926.

L. C. Porter in Cleveland

Announcement has been received that Lawrence Copeland Porter, formerly engineer in charge of special developments at the Edison Lamp Works of the General Electric Co., at Harrison, N. J., has been named manager of the health and therapeutic lamp section of the company's engineering department, with headquarters at Nela Park, Cleveland.

Mr. Porter was born in 1887 at Kensington, Conn., and received his schooling at Philadelphia, elaborating the scientific side of his training thereafter at Yale University. For more than 23 years he has been a member of the General Electric organization and has made a special study of all forms of light-projection, with special reference to automotive headlighting. He has been a Member of the Society since 1919 and served as a member of the Lighting Division of the Standards Committee from 1922 to 1927. A paper entitled, What Happens When an Automobile Headlight Is Out of Focus, by Mr. Porter and G. F. Prideaux, was published in THE JOURNAL for February, 1926, p. 222, and in TRANSACTIONS, 1926, part 1, p. 521.

Wellwood Beall, having resigned his position as aeronautic engineer with the Walter M. Murphy Co., of Pasadena, Calif., is now instructor in design at the Boeing School of Aeronautics, Oakland Municipal Airport, Oakland, Calif.

R. J. Berky, former draftsman for the Curtiss Aeroplane & Motor Co., of Garden City, N. Y., is now serving the Chance Vought Corp., of East Hartford, Conn., in a similar capacity.

Edward T. Birdsall is now engineer in the engineering and research department of the Curtiss-Wright Corp., of New York City. He was previously connected with the Wright Aeronautical Corp., of Paterson, N. J., in the same capacity.

William R. Blakely is now assistant engineer with the Huntington Aircraft Corp., of Stratford, Conn. Prior to making this connection he was a student at New York University.

A. Bruce Boehm, former service and sales manager of the Standard Oil Co. of New Jersey, at Newark, N. J., has been appointed district manager for the company at Baltimore.

Frank G. Born has become a member of the engineering department of the Delco Products Corp., of Dayton, Ohio. His previous connection was with the Wahl Co., of Chicago, for which he acted as chief engineer in charge of automotive equipment.

W. A. Britton is now occupying the position of sales and service engineer with the Continental Aircraft Engine Co., of Detroit. Previous to making this connection he was chief engineer for the Aero Engineering & Advisory Service, Inc., of New York City.

Kwang H. Chang, who has been a tool designer for Chrysler Motors, of Detroit, now occupies a similar position with the Ex-Cell-O Aircraft & Tool Corp., also of Detroit.

Gilbert W. Davis, service representative of Durant Motors, Inc., with headquarters at Kansas City, Mo., recently assumed this position after severing his connection in the same capacity with American Cirrus Engines, Inc., of Marysville, Mich.

Robert I. Dick, former engineer with the Buhl Co., of Chicago, is now assistant chief engineer for the Wisconsin Motor Co., of Milwaukee.

O. E. Eckert, who had been chief lubrication engineer for the Independent Oil & Gas Co., of Tulsa, Okla., recently accepted a position as construction engineer with the Empire Refining Co., of Bartlesville, Okla.

Frank L. Eidmann has resigned as associate professor in the School of Engineering of Princeton University to become professor of mechanical engineering at Columbia University.

Harold J. Gibson, who was a teaching assistant in mechanical engineering at the University of Michigan, recently accepted a position in the Ethyl Gasoline Corp. laboratory at Detroit.

G. O. Goller, formerly serving the General Motors G. m. b. H., at Berlin, Germany, as a body engineer, has joined the body engineering staff of the Adam Opel A. G., at Ruesselsheim on Main, Germany.

T. W. Hallerberg, for the last two years chief engineer with Lubrication Devices, Inc., of Battle Creek, Mich., is now executive engineer with the Pickwick Motor Coach Works, of Inglewood, Calif.

(*Continued on p. 42*)

Applicants Qualified

The following applicants have qualified for admission to the Society between June 10 and July 10, 1930. The various grades of membership are indicated by (M) Member; (A) Associate Member; (J) Junior; (Aff.) Affiliate; (S M) Service Member; (F M) Foreign Member.

ADAMS, CHARLES G. (A) manager, partner, Twin City Die Castings Co., 33rd and Talmadge Avenues, Southeast, *Minneapolis;* (mail) 1089 14th Avenue, Southeast.

ADAMS, WILLIAM DUDLEY (M) chief engineer, in charge of material, process and product, inspection and processing, Willard Storage Battery Co. of Canada, Ltd., 269 Campbell Avenue, *Toronto, Ont., Canada.*

ALBRIGHT, WILLIAM E. (A) chief clerk, transportation, Philadelphia Gas Works Co., 1800 North Ninth Street, *Philadelphia.*

ANDERSON, JOSEPH L. (A) designing, layout draftsman, Vimalert Co., Ltd., Garfield Avenue, Jersey City, N. J.; (mail) 1590 Williamsbridge Road, *New York City.*

ASSENMACHER, JOSEF (M) designing engineer, New Standard Aircraft Corp., 230 East 16th Street, *Paterson, N. J.*

BENZ, AUGUST (A) president, Benz Spring Co., Inc., 89-91 North Ninth Street, *Portland, Ore.;* (mail) 347 East 29th Street.

BOLAM, JOHN GEORGE (F M) motor superintendent, Anglo American Oil Co., Ltd., 36 Queen Anne's Gate, *London, S. W. 1, England.*

BOYER, GLENN C. (J) assistant engineer, specializing in powerplant and mechanical design, Burns & McDonnell Engineering Co., 406 Interstate Building, *Kansas City, Mo.*

BROOKS, M. P. (A) superintendent of equipment, State of California, Division of Highways, Equipment Department, Sacramento, Calif.; (mail) Box 381, *Bishop, Calif.*

CALHOUN, THOMAS AUSTIN (A) manager of Duplate sales, Pittsburgh Plate Glass Co., *Pittsburgh;* (mail) 1060 Morewood Avenue.

CANOOSE, JOHN E., LIEUT. (S M) United States Navy, U.S.S. Bushnell, care of Postmaster, *New York City.*

CARGO, FRANK (A) district sales representative, Bayerson Oil Works, Erie, Pa.; (mail) McCutcheon Lane, R. D. 1, Wilkinsburg Station, *Pittsburgh.*

CELLA, JOSEPH A., FIRST LIEUT. (S M) War Department, Field Artillery, City of Washington; (mail) 303 Varsity Apartments, *West Lafayette, Ind.*

CHENAULT, A. H. (M) engineer, Ethyl Gasoline Corp., 723 East Milwaukee Avenue, *Detroit.*

COE, ARTHUR B. (J) vice-president, Fort Wayne Piston Service Co., Inc., *Fort Wayne, Ind.;* (mail) 1725 Clover Lane.

COLE, CLIFFORD (A) mechanic, owner, Cole's Sales & Service, *De Witt, Mich.*

COSGROVE, JOSEPH M. (A) superintendent division No. 11, Noblitt Sparks Industries, *Columbus, Ind.;* (mail) 1642 Maple Street.

CROSET, LOUIS PAUL (F M) engineer, designer, in charge of experimental department, Aveling & Porter, Ltd., Rochester, England; (mail) Haverstock, Strangford Road, *Tankerton, Kent, England.*

CULVER, EDWIN E. (A) service manager, Jim Furlong Auto Co., *Denver, Colo.;* (mail) 57 South Pearl Street.

DALSACE, PIERRE MICHEL (J) 9 Rue Pierre Ducreux, *Paris (16e), France.*

DE GUISE, GEORGE L. (J) service manager, H. A. Bissonnette, Lowell, Mass.; (mail) 2 Chaplain Street, *Dracut, Mass.*

DELL, BRADFORD A. (A) general foreman, Cadillac Motor Car Co., *Detroit;* (mail) 2124 Liddesdale Avenue.

DEL PASO, ALEJANDRO (M) engineer, designer, Juan F. Azcárate S. en C., Colonia Federal, *Mexico, D. F., Mexico;* (mail) 2a. de Turin No. 42, *Mexico, D. F.*

DEWAR, CHARLES E. (M) factory manager, Champion Spark Plug Co., *Toledo, Ohio.*

DOTY, WADE (M) engineer, sales, Two Way Shock Absorber Co., *Jamestown, N. Y.*

ELLIS, WILBER R., FIRST LIEUT. (S M) United States Army, Coast Artillery Corps, Fort Monroe, Va.; (mail) 1298 Roosevelt Avenue, *Ann Arbor, Mich.*

EVRARD, ALPHONSE H. (M) 232 West 21st Street, *New York City.*

FIGENTZER, ERWIN THEODORE (J) parts clerk, chauffeur, Bates Chevrolet Co., Inc., *New York City;* (mail) 1171 Forest Avenue.

FORD, ARTHUR ROLAND (J) engineer, experimental department, International Motor Co., *Allentown, Pa.;* (mail) 1817½ Union Street.

GARY, CURTIS LOUIS (J) body draftsman, designer, Briggs Mfg. Co., *Detroit;* (mail) 6015 Chalmers Avenue.

GILBERT, WARREN VAN R. (M) sales engineer, Aluminum Co. of America, 3311 Dunn Road, *Detroit.*

GILLIES, HUGH A. (M) sales manager, American Brake Materials Corp., 4660 Merritt Avenue, *Detroit.*

GOTTRON, ROBERT E. (M) assistant, office of motor engineer, Cadillac Motor Car Co., *Detroit;* (mail) 8827 Steel Avenue.

GROSSPETER, FRED WILLIAM (A) 140 Hillside Terrace, *Great Kills, S. I., N. Y.*

HAMILTON, WALTER A. (M) chief engineer, Aero Corp. of California, Inc., 9401 South Western Avenue, *Los Angeles.*

HENDERSON, GEORGE F. (M) body-in-white engineer, H. H. Franklin Mfg. Co., 633 West Onondaga Street, *Syracuse, N. Y.*

HODOUS, LOUIS W. (M) plant engineer, design, operation, development of plant equipment, Canfield Oil Co., 3216 East 55th Street, *Cleveland.*

HOPKINS, GEORGE HAROLD (M) chief engineer, Vortox Mfg. Co., Claremont, Calif.; (mail) 138 North Catalina Avenue, *Pasadena, Calif.*

HORNNEY, ALVIN GEORGE (A) sales engineer, American Machine & Foundry Co., 511 Fifth Avenue, *New York City.*

HOULT, WILLIAM H. (A) superintendent of field tests, Chevrolet Motor Co., Detroit; (mail) General Motors Proving Grounds, *Milford, Mich.*

HUNT, LAWRENCE A. (J) research engineer, Ethyl Gasoline Corp., 320 Yonkers Avenue, *Yonkers, N. Y.;* (mail) P. O. Box 390.

HUTCHISON, MILLER REESE, JR. (J) independent, Hughes Building, *South Orange, N. J.*

IRVING, J. S., CAPT. (F M) technical director, chief engineer, Humber, Ltd., Coventry, England; (mail) Devonshire House, Piccadilly, *London W. 1, England.*

JACOBSON, HENRY A. (M) designing engineering department, Norton Co., New Bond Street, *Worcester, Mass.*

JANES, ARTHUR R. (M) president, general manager, Standard Foundry Co., 1600 Kewaunee Street, *Racine, Wis.*

JOHNSTON, H. (M) chief inspector, Wisconsin Axle Co., *Oshkosh, Wis.;* (mail) 142 Lake Street.

KEYS, CONRAD ROY (M) vice-president, Curtiss Aeroplane & Motor Co., Inc., *Buffalo.*

KIDD, ROY WILFRID (A) in charge of repair and overhaul shop, Aero Corp. of California, 94th and Western Avenue, *Los Angeles.*

KUNKLE, B. D. (M) president, general manager, Delco Products Corp., 329 East First Street, *Dayton, Ohio.*

LAWLER, FRANK P. (M) consulting engineer, 251 Kearny Street, *San Francisco.*

LEADEN, W. J. (J) automotive salesman, Anaconda Wire & Cable Co., New York City; (mail) 818 Fisher Building, *Detroit.*

LEEPER, ALBYN E. (A) mechanic, Studebaker Sales Co., *Chicago;* (mail) 238 North Pine Avenue.

LERCH, G. PALMER (M) mechanical supervisor, Reading Co., Sixth and Perry Streets, *Reading, Pa.*

LUNDBERG, EVERETT E. (M) chief engineer, Briggs Mfg. Co., *Detroit;* (mail) 3845 Courville Avenue.

MAHONEY, WILLIAM F. (A) commercial-car sales manager, General Motors Export Co., 1775 Broadway, Room 1528, *New York City.*

MALLOUF, NAT (M) president, Mallouf Haulage & Maintenance Corp., 10 East 40th Street, *New York City.*

MARKLAND, STANLEY (F M) engineer, in charge of research department, Leyland Motors, Ltd., Leyland, Lancashire, England; (mail) 76 Preston Road, *Chorley, Lancashire, England.*

MARTIN, LESLIE JOHN (A) superintendent, marine sales and engineering, Home Oil Distributors, Ltd., *Vancouver, British Columbia, Canada;* (mail) Capilano P. O.

MARTIN, PARKER BRADLEY (M) engineer, truck and industrial engines, Lycoming Mfg. Co., *Williamsport, Pa.;* (mail) 753 West Third Street.

MATHES, DUDLEY C. (A) general foreman, repair shop, grade 4, California Highway Commission, *Redding, Calif.;* (mail) 1008 Pine Street.

MCCAULEY, GEORGE H. (A) automotive engineer, sales department, Standard Oil Co. of New Jersey, *Baltimore;* (mail) 721 Field Street.

MCCOLLOUM, A. L. (A) sales agent, National Malleable & Steel Castings Co., 546 North Holmes Avenue, P. O. Box 185, *Indianapolis.*

MCCUTCHEON, WALTER WILLIAM (J) materials engineer, Stearman Aircraft Co., *Wichita, Kan.*

MCDOWELL, EDWARD P. (M) transportation engineer, National Dairy Products Corp., New York City; (mail) 28 Casillis Avenue, *Bronxville, N. Y.*

MICHEL, CHRIS (M) engineer, Brown-Lipe Gear Co., Syracuse, N. Y.; (mail) 1074 Salem Avenue, *Hillside, N. J.*

MILNE, ROBERT KENNETH (J) 441 Elmwood Street, North East, *Grand Rapids, Mich.*

MONAHAN, GORDON J. (A) sales manager, Canadian Raybestos Co., Ltd., Peterborough, Ont., Canada; (mail) 24 Queens Avenue, *Mimico, Ont., Canada.*

NELSON, GEORGE J. (M) factory manager, Overman Cushion Tire Co., Inc., 151 Cortlandt Street, *Belleville, N. J.*

NICHOLS, PROCTOR WALLACE (J) chief stress analyst, assistant chief engineer, Alexander Industries, Colorado Springs, Colo.; (mail) Cliff House, *Manitou, Colo.*

OLSON, GORDON C. (J) managing editor, *Motive Power,* Gillette Publishing Co., 221 East 20th Street, *Chicago.*

PEDERSEN, BRANDT (A) checker, engine and chassis, H. H. Franklin Mfg. Co., *Syracuse, N. Y.;* (mail) 1000 Bellevue Avenue, Apartment 19.

PEW, WALTER CROCKER (A) district sales manager, Philadelphia territory, Sun Oil Co., 1608 Walnut Street, *Philadelphia.*

PIKE, ROBERT D. (M) vice-president, general manager, Kalif Corp., San Francisco; (mail) 4069 Hollis Street, *Emeryville, Calif.*

POTGIETER, FRED M. (A) vice-president, engineer, Universal Gear Corp., 327 South LaSalle Street, *Chicago.*

PREST, CLARENCE O. (A) owner, manager, designing engineer, Prest Airplanes & Motors, *Arlington, Calif.*

RAUSCHER, MANFRED (M) chief engineer, Engineers' Aircraft Corp., *Stamford, Conn.*

ROGERS, CECIL RICHARD (F M) in charge of travelling service school, General Motors South African, Ltd., P. O. Box 760, *Port Elizabeth, South Africa.*

ROHRBACH, ADOLF K., DR. (F M) Rohrbach Metall-Flugzeugbau G.m.b.H., Klautschouster. 9-12, *Berlin N. 65, Germany.*

ROSE, FRANK A. (A) in charge of cars, trucks and equipment, Leo Greenbaum, *San Diego, Calif.;* (mail) 3763 Tenth Street.

Applicants for Membership

The applications for membership received between June 15 and July 15, 1930, are listed below. The members of the Society are urged to send any pertinent information with regard to those listed which the Council should have for consideration prior to their election. It is requested that such communications from members be sent promptly.

ALLEN, CHARLES W., motor tester, Packard Motor Car Co., *Detroit.*

BAHN, STEN HJORTSOE, student, International Harvester Co., *Springfield, Ohio.*

BANDEL, CHARLES A., branch sales manager, Overman Cushion Tire Co., Inc., *Baltimore.*

BERGHOFF, BREDO H., assistant chief engineer, Stalingrad Tractor Plant, *Stalingrad, Union of Socialist Soviet Republics.*

BOWEN, H. C., research engineer, Hydraulic Brake Co., *Detroit.*

BRACE, R. LYLE, lubrication technologist, sales management, National Refining Co., *Seattle, Wash.*

BROUGHTON, JAMES W., proprietor, J. W. Broughton, *Pittsburgh.*

COOK, H. WEIR, vice-president and general manager, Curtiss Wright Flying Service, *Indianapolis.*

COOKE, EDWARD FRANCIS, used car manager, General Supplies, Ltd., *Calgary, Alberta, Canada.*

COX, H. L., in charge of drafting, Canners Machinery, Ltd., *Simcoe, Ont., Canada.*

CRAIG, WALTER G., division lubricants representative and engineer, General Petroleum Corp. of California, *Los Angeles.*

DORNIER, DR. CLAUDE, Dornier Metallbaufen G.M.B.H., *Frederickshafen, Germany.*

DUBOIS, EDWARD P., laboratory assistant, aeronautical department, University of Detroit, *Detroit.*

FARRIS, CLAYTON, president, The Trucktor Corp., *Newark, N. J.*

FREUDENBERG, CARL V., layout draftsman, Servel, Inc., *Evansville, Ind.*

GOING, RICHARD F., purchasing agent, State Roads Commission, *Baltimore.*

GRAHAM, JOHN WILKINSON, lubricating engineer, Union Oil Co. of Canada, Ltd., *Vancouver, B. C., Canada.*

GREENLEAF, RALPH A., superintendent of motor vehicles, West Coast Telephone Co., *Everett, Wash.*

GUENTHER, CARLTON A., general manager, Jacobs Transfer Co. of Baltimore, *Baltimore.*

HANN, WILLIAM E., partner, Harness, Dickey, Pierce & Hann, *Detroit.*

HENDLER, FRED G., owner, Aero Pattern Works, *Buffalo.*

HUSSEY, WALTER W., service man, R. N. Wickett, Inc., *Everett, Wash.*

IRON CITY SPRING CO., INC., *Pittsburgh.*

JENSON, ROY WALLACE, in charge of parts department, J. I. Case Co., *Calgary, Alberta, Canada.*

JEWETT, FREDERIC D., chief mathematics section, Keystone Aircraft Corp., *Bristol, Pa.*

JOHNSON, W. H., service manager, Paddock Motors Co., *Everett, Wash.*

KALBFLEISCH, mechanical engineer, New Standard Aircraft Corp., *Paterson, N. J.*

KOENIG, ARTHUR CLARENCE, designer, Goodyear-Zeppelin Corp., *Akron, Ohio.*

MACFARLAN, MALCOLM SCHORR, draftsman, The Celotex Co., *Marrero, La.*

MCFARLAND, ERNEST R., service manager, Imperial Motors, Ltd., *Calgary, Alberta, Canada.*

MCNEILL, MAURICE R., manager truck and bus tire department, Firestone Tire & Rubber Co. of California, Portland Branch, *Portland, Ore.*

MEISTER, REGINALD E. B., engineer, Murray Motor Car Co., *Everett, Mass.*

MILLER, SCOTT L., assistant superintendent of maintenance, Consolidated Coach Co., *Lexington, Ky.*

MILLER, WELLS B., district service manager, The White Co., *Baltimore.*

O'DONNELL, JAMES H., service manager, Mack International Motor Truck Corp., *Baltimore.*

ORMSBY, EDMUND B., detail draftsman, aeronautical, Naval Aircraft Factory, *Philadelphia.*

PETERSON, WALTER THOMAS, manager, sales, Penn Rivet Corp., *Detroit.*

SANG, HENRY JAMES, junior engineer, Naval Aircraft Factory, *Philadelphia.*

SARDOU, CHARLES, JR., engineer, Stearman Aircraft Co., *Wichita, Kan.*

SCHULTZ, EDWARD W., shop foreman, R. N. Wickett, Inc., *Everett, Wash.*

SISK, PAUL H., engineer in charge of design and production, Wichita Falls Motor Co., *Wichita Falls, Tex.*

SOCIETE ANONYME DES ACIERIES CI-DEVANT, Georges Fischer, *Schaffhouse, Switzerland.*

STIVERS, HARRY LEE, division motor-vehicle superintendent, Bell Telephone Co. of Pennsylvania, *Wilkes-Barre, Pa.*

STRONG, THOMAS L., junior mechanical engineer, United States Air Corps, Wright Field, *Dayton, Ohio.*

STROUSE, JAY S., manager, General Motors Truck Co., *Baltimore.*

TIMOSHENKO, STEPHEN, professor of engineering mechanics, University of Michigan, *Ann Arbor, Mich.*

VAIL, RICHARD, editor, Autobody Publishing Co., Inc., *New York City.*

WENZEL, C. LEO, manager truck and bus wheel department, Budd Wheel Co., *Detroit.*

WHITE, V. K., assistant to director of service, Auburn Automobile Co., *Auburn, Ind.*

WINN, A. F., traffic manager and manager aviation sales, Skelly Oil Co., *Tulsa, Okla.*

WILLIAMSON, GEORGE, manager and director, Hay Motor & Engineering, Proprietary, Ltd., *Hay, New South Wales, Australia.*

ROSENKRANZ, J. A. (A) president, National Automotive & Electrical School, Figueroa and Santa Barbara Street, *Los Angeles.*

ROSS, FRED W. (F M) engineer, Euston Ignition Co., London, England; (mail) 60 Hartington Street, *Barrow in Furness, Lancashire, England.*

RUIZ, EDWARD A. (A) in charge of export billing and consular service, Brockway Motor Truck Corp., Export Department, New York City; (mail) 304 Park Place, *Brooklyn, N. Y.*

SALKIN, M. (J) draftsman, Oil Well Supply Co., *Oil City, Pa.;* (mail) 1003 East Second Street.

SCHAMEHORN, A. J. (M) director, General Motors Corp. Proving Grounds, *Milford, Mich.*

SCHECHTER, BEN (M) chief engineer, American Car Co., 1558 South Vandeventer Avenue, *St. Louis.*

SCHILDHAUER, CLARENCE H. (M) assistant to vice-president and general manager, Dornier Co. of America, 1775 Broadway, *New York City.*

SHELTON, WILLIAM J. (M) experimental department, Timken-Detroit Axle Co., *Detroit.*

SHUTTS, O. M. (M) designing engineer, Lycoming Mfg. Co., *Williamsport, Pa.*

SIMONS, OTMAR F. (A) specification engineer, Gramm Motors, Inc., Delphos, Ohio; (mail) 1139 West Market Street, *Lima, Ohio.*

SPANGLER, FRANK (M) designer, checker, General Motors Radio Corp., *Dayton, Ohio;* (mail) 740 Edgemont Avenue.

SPOHR, CARL W. (M) chief draftsman, Vacturi Carburetor Co., 1253 Diversey Parkway, *Chicago.*

STAFFORD, ROBERT LESLIE (F M) sales engineer, technical assistant to director, W. H. Dorman & Co., Ltd., *Stafford, England.*

STEINKAMP, WILBERT H. (J) sales engineer, representative, Wright Aeronautical Corp., Paterson, N. J.; (mail) 89 Hyatt Avenue, *Yonkers, N. Y.*

STEWART, CHARLES R. (M) development engineer, Firestone Tire & Rubber Co., *Los Angeles.*

SWANCAR, JOSEPH EDWARD (A) head of commercial division of Canada, Ford Motor Co. of Canada, Ltd., East Windsor, Ont., Canada; (mail) 68 Prado Place, *Riverside, Ont., Canada.*

TEPPER, FRANK E. (M) division supervisor of motor vehicles, American Telephone & Telegraph Co., *Philadelphia;* (mail) 406 Morton Avenue, Ridley Park.

THOMAS, FREDERICK METCALF (J) assistant aeronautical engineer, Pratt & Whitney Aircraft Co., *East Hartford, Conn.*

TROLLOPE, CHARLES HENRY (A) service representative, H. B. Phipps, 58 Margaret Street, *Sydney, New South Wales, Australia.*

VAN CAMP, BENJAMIN T. (A) transportation engineer, George A. Hormel & Co., *Austin, Minn.*

VANCURA, EMIL (A) president, manager, Vancura Machine Co., Inc., Northern Boulevard and 33rd Street, *Long Island City, N. Y.*

VOLLHEIM, WILLIAM (J) draftsman, Fairbanks, Morse & Co., *Beloit, Wis.;* (mail) 717 Parker Avenue.

VON MERTENS, ERNST (J) draftsman, designer, Robert Bosch Magneto Co., Inc., Long Island City, N. Y.; (mail) 139 West 75th Street, *New York City.*

WADDILL, WALTON (J) assistant in experimental testing laboratory, International Motor Co., Allentown, Pa.; (mail) Industrial Y.M.C.A., *Bayonne, N. J.*

WEISS, CARL W. (M) chief engineer, vice-president, Weiss Engineering Corp., New York City; (mail) 1050 Ocean Avenue, *Brooklyn, N. Y.*

WELLS, CHESTER R. (M) engineer, in charge of special development, brakes and fuel feeds, Stewart-Warner Corp., *Chicago;* (mail) 2635 Carmen Avenue.

WHEATON, ABRAM W. (A) president, A. W. Wheaton Brass Works, 110 Walnut Street, *Newark, N. J.*

WHEELER, P. R. (J) junior aeronautical engineer, Arlington engine-test laboratory, Bureau of Standards, City of Washington; (mail) 3 Cedar Street, Mount Ida, *Alexandria, Va.*

WHITE, W. KING (A) president, treasurer, Cleveland Tractor Co., 19300 Euclid Avenue, *Cleveland.*

WICK, OSCAR (A) president, C. W. S. Grinding & Machine Works, *Portland, Ore.;* (mail) 443 Burnside Street.

WILKINSON, GEORGE SHAKESPEARE (F M) chief engineer, D. Napier & Son, Ltd., Acton, London, England; (mail) Marden Ash, Hillcrest Road, Ealing, *London W. 5, England.*

WILLIAMS, C. E. (M) carbureter engineer, Cadillac Motor Car Co., *Detroit;* (mail) 2956 Pasadena.

WINSTON, A. W. (M) experimental engineer, in charge of fabrication development of magnesium alloys, Dow Chemical Co., *Midland, Mich.*

WOLLERING, ELMORE F. (J) constructional development tire engineer, United States Rubber Co., *Detroit;* (mail) 36 McLean Avenue.

WOOD, DONALD B. (M) sales engineer, General Motors South African, Ltd., *Port Elizabeth, Rhodesia, Africa.*

Notes and Reviews

AIRCRAFT

These items, which are prepared by the Research Department, give brief descriptions of technical books and articles on automotive subjects. As a general rule, no attempt is made to give an exhaustive review, the purpose being to indicate what of special interest to the automotive industry has been published.

The letters and numbers in brackets following the titles classify the articles into the following divisions and subdivisions: *Divisions*—A, Aircraft; B, Body; C, Chassis Parts; D, Education; E, Engines; F, Highways; G, Material; H, Miscellaneous; I, Motorboat; J, Motorcoach; K, Motor-Truck; L, Passenger Car; M, Tractor. *Subdivisions*—1, Design and Research; 2, Maintenance and Service; 3, Miscellaneous; 4, Operation; 5, Production; 6, Sales.

The Design of Airplane Wing Ribs. By J. A. Newlin and George W. Trayer. Report No. 345. Published by the National Advisory Committee for Aeronautics, City of Washington, 1930; 54 pp., illustrated. [A-1]

In obtaining information for the design of truss and plywood forms, tests were made at the Forest Products Laboratory on many designs of wing ribs. Different types in various sizes were compared by means of these tests, in accordance with the request of the Bureau of Aeronautics of the Navy Department. Many tests were also made on parallel-chord specimens of truss and plywood forms and on parts of wing ribs, such as truss diagonals and sections of cap strips. It was found that, for ribs of any size or proportions, when designed to obtain a well-balanced construction and carefully manufactured, distinct types of truss and forms are of various efficiencies. With ideal construction the truss comes first; second, a lightened and reinforced plywood type; third, a full-plywood web type with stiffeners; fourth, a plywood web with lightening holes and no reinforcing; and, fifth, a full web with no stiffeners.

Each type has its place in airplane design because manufacturing difficulties set up practical limits for the various types. For example, shallow trusses cannot be manufactured and assembled without great difficulty. Neither can a reinforced plywood truss be substituted for a full-plywood type when to obtain maximum efficiency an excessively thin plywood must be used.

The results of the investigation are intended to warn the designer against faulty construction and to aid him in formulating rules regarding the effect of various factors on the design and strength of different parts.

A Method of Calculating the Ultimate Strength of Continuous Beams. By J. A. Newlin and George W. Trayer. Report No. 347. Published by the National Advisory Committee for Aeronautics, City of Washington, 1930; 28 pp., illustrated. [A-1]

In the design of continuous beams subjected only to transverse load, the common practice has been to estimate the maximum loads by substituting the numerical value of the modulus of rupture, as obtained in bending tests, in the usual equation of three moments. Further, for combined axial and transverse loading, two methods of calculation have been used. The more common one is the application of the generalized equation of three moments, while the other is an extension of the ordinary equation of three moments to allow for the moments introduced by the direct tension or compression load. In the second method, the deflection in the span at the point of maximum moment is calculated, neglecting the effect of axial load, and the product of the axial load and this deflection is added to the moment determined by the ordinary equation of three moments.

Both of these methods are used to calculate maximum loads, although neither is properly applicable beyond the elastic limit. The purpose of the study reported here was to investigate conditions after the elastic limit has been passed. As a result of the study, a method of calculation that is applicable to maximum load conditions has been developed. The method is simpler than those now in use and applies properly to conditions where the present methods fail to apply.

The experimental work was conducted at the Forest Products Laboratory in cooperation with the Bureau of Aeronautics of the Navy Department, and the report was submitted to the National Advisory Committee for Aeronautics for publication. More than 300 continuous beams were tested under transverse load and under combined axial and transverse load. Loads obtained in test for beams of rectangular section were as much as 50 per cent in excess of loads calculated by the usual methods, with the average excess about 25 per cent. For I-beams the average increase was about 40 per cent. Fortunately, the error in the usual calculation is on the side of safety, but it is too great to be neglected in good design.

A Possible Method for Preventing the Autorotation of Airplane Wings. By Oskar Schrenk. Translated from *Zeitschrift für Flugtechnik und Motorluftschiffahrt*, Nov. 14, 1929. Technical Memorandum No. 569; 6 pp., 9 figures. [A-1]

Gloster Metal Construction. Reprinted from *Flight*, April 18, 1930. Technical Memorandum No. 570; 9 pp., 26 figures. [A-1]

The foregoing Technical Memoranda were issued during June by the National Advisory Committee for Aeronautics, City of Washington.

Aircraft-Float Design and Construction. By Holden C. Richardson. *Transactions of the American Society of Mechanical Engineers, Aeronautical Engineering Section*, April-June, 1930, p. 81. [A-1]

Designs of aircraft floats have included the canoe type, inflated bag or skid, box type, multi-rib type, canoe system with cloth over a thin smooth skin, girder type, and all-metal construction. The author follows the development and discusses necessary characteristics. Late construction is principally of aluminum alloy for both floats and hulls. Bulkheads are used to carry concentrated loads, and the bottoms are sometimes reinforced by exterior longitudinals. Extruded shapes, angles, channels, and tees are becoming popular as framing members.

Le Soudure Autogène dans les Constructions Aéronautiques. By R. Granjon. Published in *L'Aéronautique*, May, 1930, p. 169. [A-1]

The welding of aircraft, introduced in France during the World War, found in some of its friends there its worst enemies. Their over-enthusiasm led them to apply the process beyond its field of usefulness and beyond their own knowledge. These abortive attempts resulted in failure of the aircraft and in accidents, as a natural consequence of which welding fell into disrepute.

Two years ago the Office Central de la Soudure, a non-commercial organization devoted to research in the field of welding, took up the cudgels on a more scientific basis and negotiated with the Services Techniques a sale of two single-engine and one tri-motor six-seated aircraft, with the proviso that the welded fuselage should successfully pass static tests. The completed fuselage, tested for a factor of 5, failed only at 6.5.

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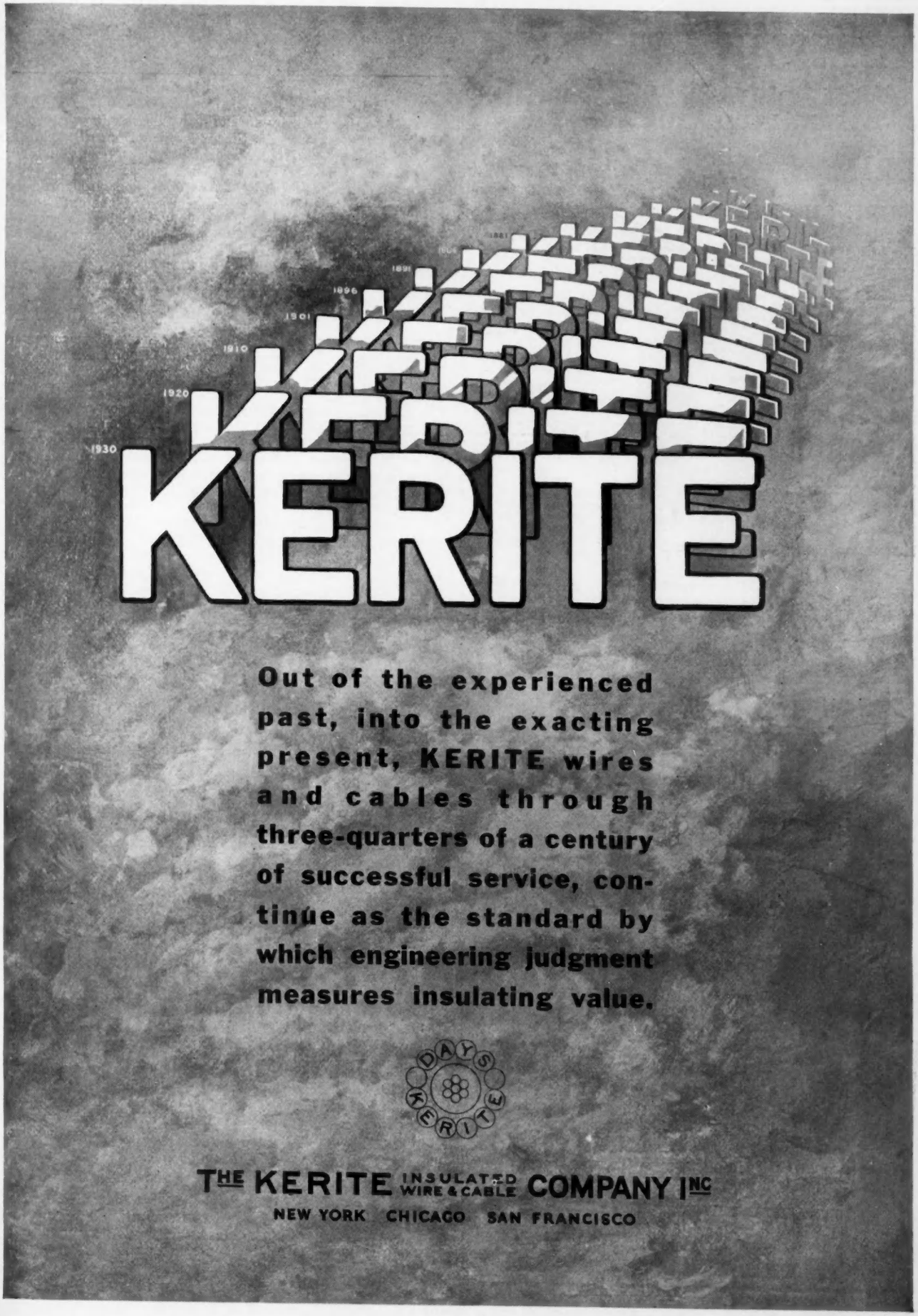
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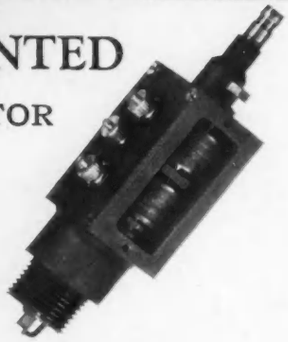
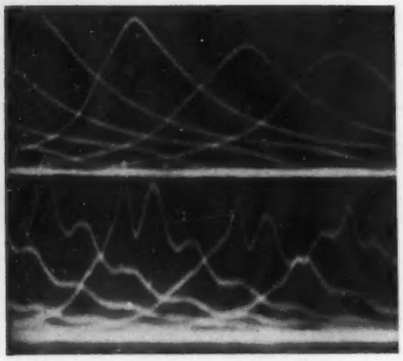




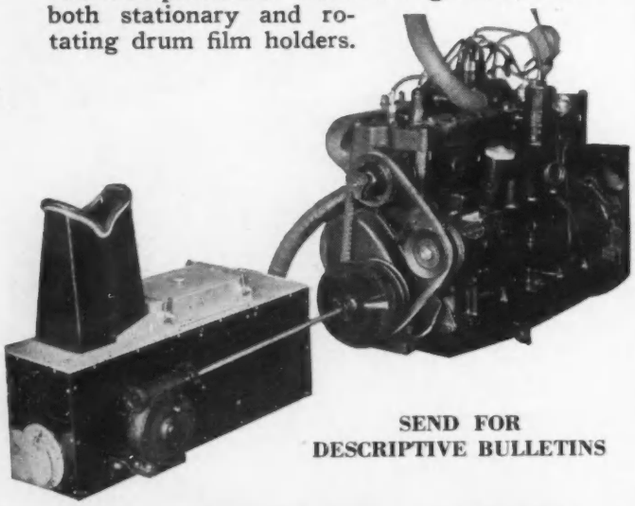
Curves taken with flush mounted Electric Engine Indicator. Note smoothness of wave form.

This indicator is sold exclusively by Southwark Foundry and Machine Company and Westinghouse Electric and Manufacturing Company, under licenses from General Motors Corporation and Electrical Research Products, Inc.

This diagram shows surges occurring when the same indicator is mounted on the end of a short tube.

Tube effect, clearly shown above, is a very appreciable obstacle in recording true values. The McCollum-Peters Electric Engine Indicator, designed for flush mounting, shows a distorsionless standing wave form with tube surge eliminated.

Below is shown a Multiple Element Oscillograph direct coupled to Engine Shaft for absolute synchronism. Equipment furnished includes rotating mirror, viewing screen for visual inspection of the standing wave, also both stationary and rotating drum film holders.



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Notes and Reviews

Continued

The present article, written by the director of the Office Centrale, is designed, by spreading technical knowledge, to prevent a recurrence of the misapplication of welding through ignorance. He deplores the over-emphasis put on the human element in welding, and argues that a well-founded technique and a regulated and supervised production routine will nullify the importance of the individual welder. Such technique should include the choice of metals and the design of the welded parts, which points he discusses, inveighing particularly against the tendency to prize to the exclusion of other factors the appearance of the weld. His text, summarized, is that the engineer, not the operator, governs the success of welding.

..

Über die Längsstabilität eines Flugzeuges mit losgelassenem Höhensteuer. By Hermann Blenk. Published in *Zeitschrift für Flugtechnik und Motorluftschiffahrt*, April, 1930, p. 189. [A-1]

In Germany every airplane is required to possess stability in three directions with released controls. Actual tests with individual craft have proved that this condition can be met, the author asserts, but no thorough investigation of the factors on which stability depends has been made. To fill this gap for the case of longitudinal stability is the purpose of the present article.

The treatment is theoretical and formulas are presented expressing the forces present in pitching, the pressure on the horizontal tail surfaces, conditions for stability with controlled and uncontrolled elevator and for certain special cases. Only gliding flight is considered; a subsequent article dealing with power flight is projected, and after that a checking of the theory here formulated with the results of actual tests of aircraft.

Theorie des Landestosses von Seeflugzeugen. By Wilhelm Pabst. Published in *Zeitschrift für Flugtechnik und Motorluftschiffahrt*, May 14, 1930, p. 217. [A-1]

This theoretical investigation into the impact loading of seaplanes when alighting on and starting from water is intended to bring out more clearly than has been done before the influence of various factors, to evaluate the significance of results arrived at empirically and so to furnish the design fundamentals for the building of pontoons and flying boats. The methods here followed are based on a physical explanation of landing shocks previously written by F. Seewald, editor of the aerodynamic department of the German Institute for Aeronautical Research.

In enumerating the factors connected with the craft itself that influence landing shocks, the author points out that, in addition to those usually considered, such as the weight and speed of the plane and the size and shape of the pontoon, the elasticity of the structure is of the utmost importance. In his opinion the influence of the damping action of the material should also be investigated. Of prime significance, he points out further, is the extent of the area of the pontoons that comes at any one time into contact with the water. This depends not only on the design of the pontoons but on the type of landing made and the condition of the water. So many different conditions and combinations of conditions may be introduced by the last two factors that, for the purpose of this investigation, certain conventional simplifications are adopted which make possible arriving at qualitative if not quantitative results and the comparison of different designs.

After developing his formulas, the author applies them to a Heinkel monoplane flying boat and compares the results obtained by him with the loading allowances of the German Institute for Aeronautical Research. Experiments are now being carried out to check the theory presented and these will be reported on in the near future.

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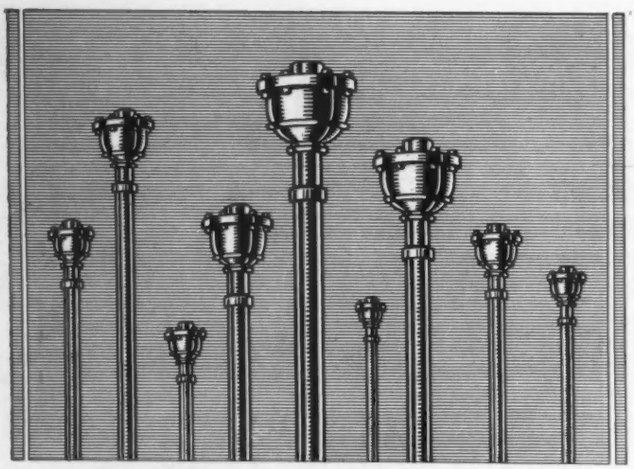

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Notes and Reviews

Continued

Le Radiophare de Bobigny pour la Navigation Aérienne. By Marcel Cleriot. Published in *L'Aéronautique*, May, p. 174, and June, 1930, p. 221. [A-1]

Radio beacons, although comparatively old in the United States, are more in the nature of innovations overseas. In fact, the beacon at Bobigny, France, is said to constitute the first application of this aid to aerial navigation in Europe. It was erected near the Bourget airport and has been in successful operation for eight months, guiding the pilots of the Air Union in their flights between Paris and London and Paris and Marseilles. It has just been purchased by the French Air Ministry.

The general principles governing the action of radio beacons are discussed in detail, and the construction of the particular beacon under consideration, the aircraft receiving sets and the method of its operation are described. Advantages claimed for the beacon are simplicity of principle, operation and use; the employment of a long wave-length; and the fact that it necessitates no extra equipment on the aircraft.

Service Monographs of the United States Government, No. 61. Aeronautics Branch, Department of Commerce. Published by The Brookings Institute, Washington, D. C., 1930; 147 pages. Price $1.50. [A-3]

These monographs are all prepared according to a uniform plan. They give, first, the history of the establishment and development of the service; second, its functions, described, not in general terms, but by detailing its specific activities; third, its organization for the handling of these activities; fourth, the character of its plant; fifth, a compilation of, or reference to, the laws and regulations governing its operations; sixth, financial statements showing its appropriations, expenditures and other data for a period of years; and, finally, a full bibliography of the sources of information, official and private, bearing on the service and its operations.

These studies are wholly descriptive in character. No attempt is made in them to subject the conditions described to criticisms nor to indicate features in respect to which changes might with advantage be made.

The monographs reveal the extent to which work in the same field is being performed by different services. Through them it will also be possible to subject any particular feature of the administrative work of the Government to exhaustive study.

The monographs will serve the double purpose of furnishing an essential tool for efficient legislation, administration and popular control, and of laying the basis for critical and constructive work on the part of those upon whom responsibility for such work primarily rests.

ENGINES

Performance of a High-Speed Compression-Ignition Engine Using Multiple-Orifice Fuel-Injection Nozzles. By J. A. Spanogle and H. H. Foster. Technical Note No. 344. Published by the National Advisory Committee for Aeronautics, City of Washington, 1930; 18 pp., 25 figures. [E-1]

This report presents test results obtained during an investigation to determine the relative performance of a single-cylinder, high-speed, compression-ignition engine when using fuel-injection-valve nozzles with different numbers, sizes and directions of round orifice. A spring-loaded automatic injection-valve was used, centrally located at the top of a vertical disc-type combustion-chamber formed between horizontally opposed inlet and exhaust valves of a 5 x 7-in. engine.

A series of fuel-injection-valve nozzles with different arrangements of round orifices were tested, starting with

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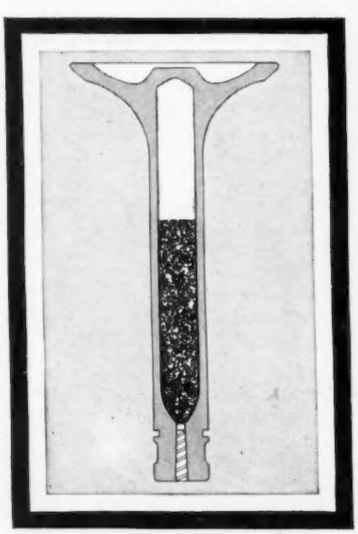

Section of Rich Salt-Cooled Valve showing salt mixture in the valve stem. After motor reaches operating temperature, salts melt and rapid motion of the valves circulates the solution, carrying heat away from valve seat. Salt-cooling gives four important advantages: (1) Valves run "black" under conditions where un-cooled valves operate at "red" heats. (2) Lessened tendency toward detonation and pre-ignition, because of lower operating temperatures. (3) Elimination of valve burning, distortion, pitting and wear. (4) Improves the fuel consumption by lessening the tendency for detonation with lean mixtures.

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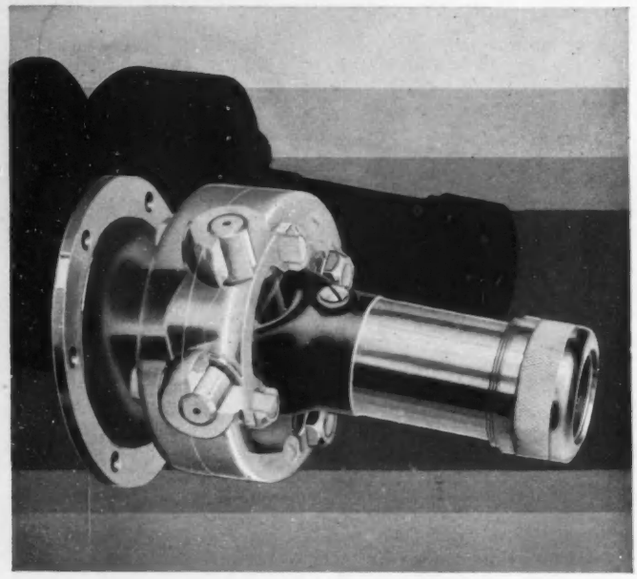

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Notes and Reviews

Continued

orifices so small that impingement on the combustion-chamber walls was impossible and increasing beyond the start of impingement. From these data two main jets of 0.018-in. diameter were decided upon, and the other jets varied in number, size and direction. A nozzle was tested with the area of the orifices proportional to the volume of air to be served by the jets using the two 0.018-in. orifices as a basis.

A table and curves showing the performance of the engine with different nozzles are presented and indicator cards and spray photographs are included. The test results are discussed, and some probable reasons are given for the variation in performance with different nozzles on a basis of spray distribution. Conclusions are drawn as to the possible application of these results to the design of fuel-injection-valve nozzles for use with a combustion-chamber having a low rate of air-flow.

Pressure Fluctuations in a Common-Rail Fuel-Injection System. By A. M. Rothrock. Paper presented at the Third National Oil and Gas Power Meeting, State College, Pa., June 12 to 14, 1930. [E-1]

The instantaneous pressures at the discharge orifice of a common-rail fuel-injection system were determined by analyzing the stem-lift records of an automatic injection-valve. The fuel injection was obtained by releasing fuel under high pressure from a reservoir by means of a cam-operated timing-valve. The period of injection was controlled by the opening of a second cam-operated valve which reduced the fuel pressure between the timing valve and the injection valve to atmospheric pressure.

The results show that pressure-wave phenomena occur between the high-pressure reservoir and the injection valve, but that these pressure waves can be controlled in a manner advantageous to the injection of the fuel. The results also give data as to the design of such an injection system for a high-speed fuel-injection engine. An injection system of this type assures the same rate of fuel discharge regardless of engine speed.

Der Fahrzeug-Dieselmotor der Linke-Hofmann-Busch-Werke A.-G. Breslau. By Ludwig Hausfelder. Published in *Automobiltechnische Zeitschrift*, June 10, 1930, p. 392. [E-1]

Among all the newer German automotive Diesel engines, the author states, the Linke is distinguished by a number of meritorious construction features. It belongs to the solid-injection four-stroke-cycle class, and in the present production schedules are included engines of 50, 85, 150, 300 and 400-hp. output at speeds ranging from 1000 to 1200 r.p.m. Their weight ranges from about 15 lb. to 26 lb. per hp., depending on whether light alloys or cast iron is used in their construction.

The construction and operation of the smallest type, designed for automotive use, are described. This type includes a four-cylinder 115 x 165-mm. (4.5 x 6.5-in.) engine with an output of 50 hp. and a six-cylinder 120 x 170-mm. (4.7 x 6.7-in.) engine with an output of 85 hp. at 1200 r.p.m.

Die Berechnung einer Zweitakt-Dieselmaschine. By Dr. Walter Haeder. Published by Richard Carl Schmidt & Co., Berlin, Germany. 380 pp.; 237 illustrations. [E-1]

This handbook is intended to serve as a guide and source of reference for both the study and the practical design of Diesel engines. The design of a 400-hp. two-stroke-cycle solid-injection Diesel engine is described in detail. Each separate part is worked out completely, and the important accessories, such as the fuel, lubricating and water pumps, radiator and reverse gear, are similarly treated. The necessary calculations and working drawings are included in the treatise.

(*Continued on next left-hand page*)

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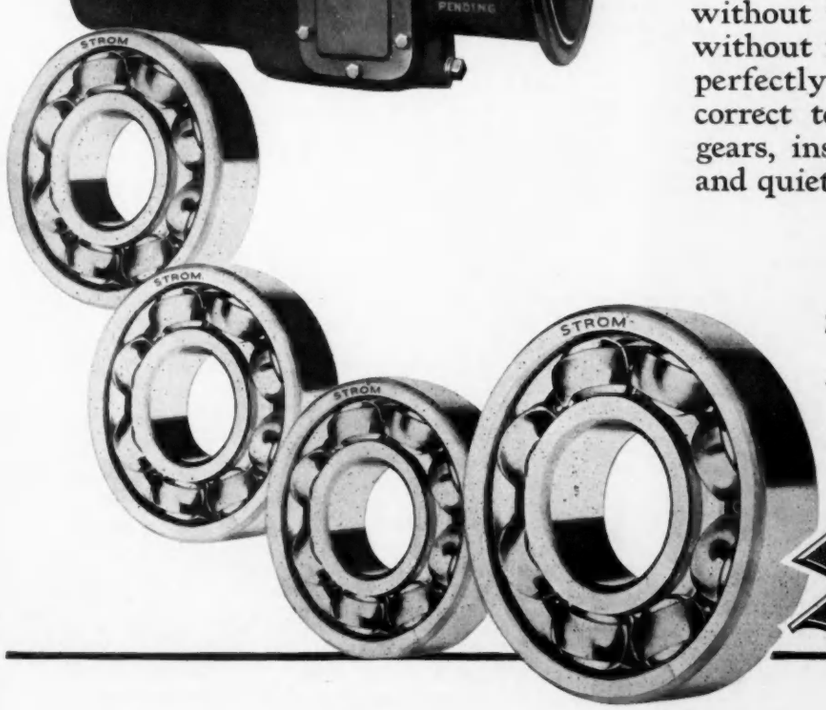

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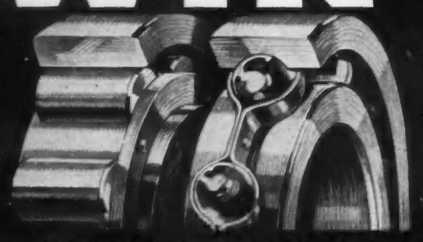
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Notes and Reviews

Continued

While the author has aimed to keep his work sufficiently simple to be easily understood, he has included enough explanatory and fundamental material to enable the methods presented for the design of the specific engine described to be applied to other Diesel types.

Introducing the more practical portion of the book is a discussion of internal-combustion engines, the advantages and disadvantages of the solid-injection Diesel, the relative merits of the two and the four-stroke-cycle and the Diesel combustion processes.

A Study on Airplane Ceiling. By Louis Breguet and Maurice Roy. *Transactions of the American Society of Mechanical Engineers, Aeronautical Engineering Section,* April-June, 1930, p. 61. [E-1]

The useful torque exerted on the propeller-shaft depends principally on the altitude, the throttle setting, and the r.p.m. of the engine. While the French Section Technique d'Aeronautique postulates that the variation of torque with altitude depends mainly on the air density, the authors believe that, given a good altitude adjustment and a carbureter heated to a constant temperature, the variation of torque depends directly on the relative pressure at altitude.

Having established formulas for power variation with altitude, it is possible to calculate the ceiling theoretically, provided the characteristics of the plane have been determind by tests in horizontal flight at any altitude and at any load whatsoever.

With a specially designed plane and an unsupercharged engine, the authors estimate that an altitude of 13,000 meters (42,640 ft.) can be reached, beating previous records by some 2000 meters (6560 ft.).

A few simple tests at one altitude with a careful record of all the elements involved can be made to give all the plane and propulsive-group characteristics. The theory and practice of such experiments are fully developed in the paper.

MATERIALS

A Survey of the Sulphur Content of Commercial Motor Fuels, 1930. By A. J. Kraemer. Mimeographed Report No. 3026, issued by the United States Bureau of Mines, Government Printing Office, City of Washington, 1930; 11 pp. [G-1]

During the last few years a great deal of discussion has centered around the question of the permissible limit of sulphur content in fuels for automotive and aviation engines. Under certain conditions of operation, sulphur in motor fuels causes the formation of corrosive substances which attack bearings, wristpins and other parts of engines. On the other hand, the author points out that, in making gasoline from certain stocks, the gasoline fraction normally contains a relatively large percentage of sulphur compounds. By the generally used processes of reducing the sulphur content, a significant part of the gasoline fraction is lost as motor fuel, and added expenses are entailed for chemicals, labor and other process costs. This condition has been an economic burden to some refiners of high-sulphur crudes.

The data presented in this paper indicate that the reduction of sulphur content of motor fuel is a problem to a relatively small number of refiners and of little or no concern to the remainder; in fact, more than 90 per cent of the 153 samples analyzed passed the Federal specification requirement, which limits the sulphur content to 0.10 per cent.

This report is based upon the results of an examination for sulphur content, Doctor test, copper-strip corrosion, and color of the samples collected during Jaunary, 1930, for the 21st semi-annual gasoline survey of the Bureau of Mines.

(*Continued on next left-hand page*)

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Notes and Reviews

Continued

The Significance of Tests for Motor Fuels. By R. E. Wilson. Paper presented at the Detroit Regional Meeting, American Society for Testing Materials, March, 1930. [G-1]

In this paper the author discusses the significance of the various tests for motor fuels, particularly in the light of extensive research along these lines during the last few years by various industrial laboratories and the Bureau of Standards.

Volatility, the importance of the 10, 50 and 90-per cent points of a fuel, antiknock quality, sulphur and gum content, and freedom from corrosive substances are discussed and their significance indicated.

The author points out that high volatility and antiknock properties and low sulphur and gum content all cost money to obtain, and that frequently the automotive engineer can, by proper design, decrease the need for too rigid specifications on all these points. Chief among the suggestions to the automotive engineer are: (*a*) more care in design to prevent overheating of fuel lines, vacuum tank, and so forth, thus permitting the use of more volatile fuels without danger of vapor lock; (*b*) preventing crankcase corrosion and thus raising permissible limits for sulphur by eliminating the condensation of water in engines and crankcases; and (*c*) more uniform cooling of engines, especially aircraft engines, to reduce the tendency of hydrocarbon fuels to detonate.

A Study of Slip Lines, Strain Lines and Cracks in Metals under Repeated Stress. By Herbert F. Moore and Tibor Ver. Bulletin No. 208. Published by the Engineering Experiment Station, University of Illinois, Urbana, Ill., 1930; 60 pp..; illustrated. [G-1]

Much uncertainty still surrounds the mechanism of the fracture of metals under stress, and the interrelation and significance of certain signs of structural damage to metals furnish fruitful fields of study.

At the University of Technical Sciences at Budapest Dr. Ver had carried on a study of the development of slip lines, strain lines and cracks in metal subjected to repeated stress. When he came to the University of Illinois as holder of the Jeremiah Smith Scholarship from the Hungarian Board of Education, he planned to continue this research. His work was done under the general direction of Professor Moore, in charge of the fatigue-of-metals laboratory. The metals studied were: Armco iron, S.A.E. 1020 plain carbon steel, S.A.E. 3135 chromium-nickel steel, stainless iron, brass, monel metal and duralumin.

The report describes the apparatus used and the methods of test, and gives a comprehensive set of conclusions. It is liberally illustrated with micrographs of the various metals tested.

Identification of Aircraft Tubing by Rockwell Test. By Horace Knerr. Technical Note No. 342; 8 pp. illustrated [G-1]

Strength in Shear of Thin Curved Sheets of Alclad. By George Michael Smith. Technical Note No. 343; 27 pp., 10 figures. [G-1]

The foregoing two Technical Notes were published by the National Advisory Committee for Aeronautics, City of Washington, during June.

Light Metals in the Automotive Industry. By Zay Jeffries. Paper presented at the Detroit Regional Meeting, American Society for Testing Materials, March, 1930. [G-1]

This paper discusses the properties and uses of American alloys. Many of the tabular data were previously presented in a paper prepared for the World Engineering Congress which was abstracted in the December, 1929, issue of *Metals and Alloys*.

(*Continued on second left-hand page*)

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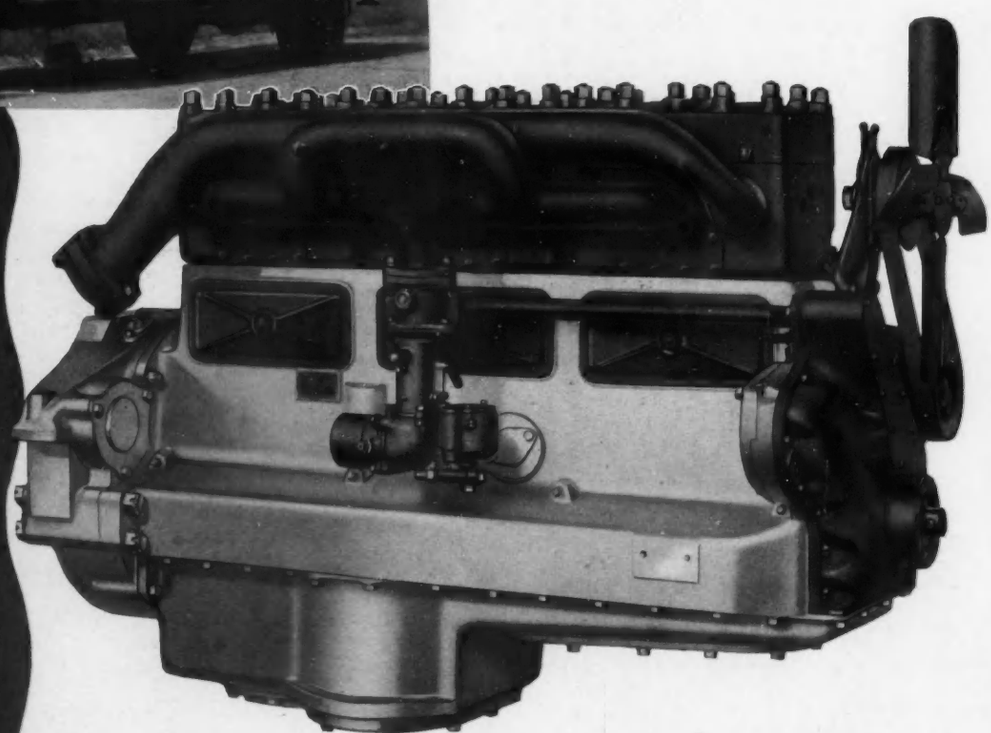
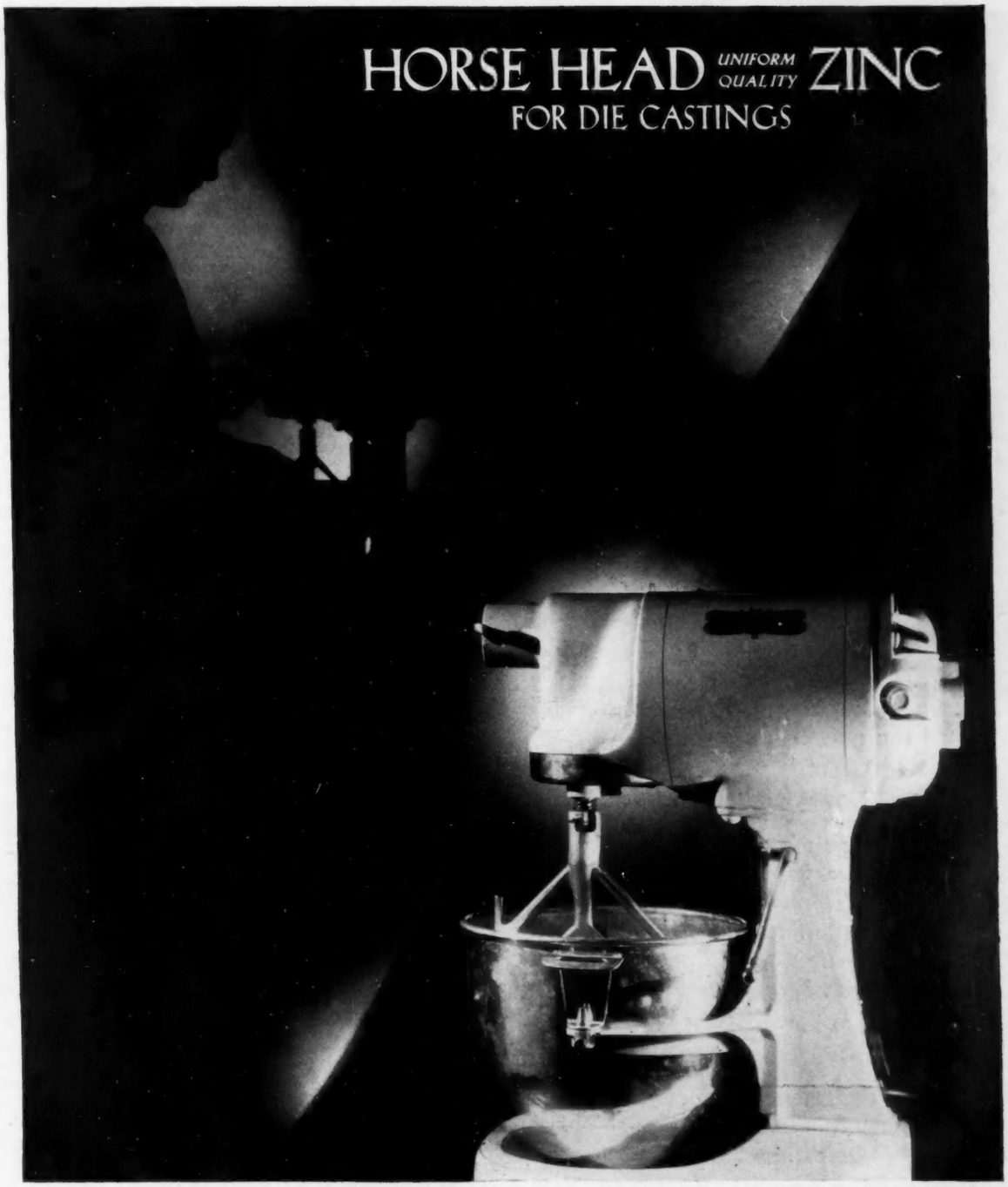


WAUKESHA ENGINES



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Notes and Reviews

Continued

The Influence of Stress Range and Cycle Frequency on Corrosion. By D. J. McAdam, Jr. Paper presented at the Annual Meeting of the American Society for Testing Materials, Atlantic City, June 23-27, 1930. [G-1]

Part I of the paper gives a brief résumé of previous work, an outline of continued investigation and a description of methods and material. Part II discusses the influence of cyclic stress on corrosion, and stressless corrosion, total damage and net damage are illustrated. Parts III, IV and V discuss the influence of stress range and cycle frequency on corrosion of steels, aluminum alloys range and cycle frequency on the rate of net damage, and Part VII is a general discussion of the influence of cyclic and monel metal. Part VI discusses the influence of stress stress on corrosion.

Les Récents Progrès des Alliages Légers d'Aluminium. By Léon Guillet. Published in *Le Génie Civil*, May 3, p. 427; May 10, p. 449; May 17, 1930, p. 475. [G-1]

Alpax, on the ground of its unusually useful mechanical properties and physical characteristics, is singled out as of particular importance in this story of recent aluminum-alloy exploitation. Alpax is praised for its castability, low density, resistance to atmospheric and chemical agents (of the same order as that of 99.5-per cent pure aluminum) and its usefulness in the following industries: aeronautic, automotive, railroad, chemical, electrical, mechanical, mineral and optical.

This article, written by the director of the l'École Centrale des Arts et Manufactures, takes up the story of aluminum alloys where it was left in 1924 by a thorough review in this same magazine, and presents a summary of the scientific and industrial development in the succeeding six years.

For the purpose of the article, aluminum alloys are defined as those containing between 80 and 90 per cent aluminum, the density of which varies from 2.6 to 3.2, and within this classification are included the following types: aluminum-silicon, aluminum-silicon-magnesium, aluminum-copper, aluminum-copper-silicon, aluminum-copper-silicon-magnesium (with or without nickel, zinc or manganese), alloys including cadmium and zinc, and other miscellaneous alloys. The commercial alloys discussed include Telectal; LM, of the Alais, Froges & Camargue Co., Lautal; duralumin; and RR 50, which is extensively used by the Rolls-Royce company and Scleron. A particularly interesting development is said to be Vetal, a bi-metal analogous to Alclad, having a duralumin base and a protective covering of pure aluminum.

Important articles and investigations dealing with the alloys are summarized, the aspects treated including their physical characteristics and chemical properties, their mechanical treatment and heat-treatment and their industrial utilization.

Aluminum and Its Production. By Junius David Edwards, Francis C. Frary and Zay Jeffries, with the collaboration of a group of other experts from the staff of the Aluminum Co. of America. Published by the McGraw-Hill Book Co., New York City and London, 1930; 2 volumes, Vol. 1. 358 pp.; Vol. 2, 870 pp., indexed and illustrated. Price $12 per set. [G-5]

Good technical literature on aluminum and its production is scarce. Furthermore, most of it is very much out of date. Therefore the issuance of a two-volume work which is a veritable library on the subject, as one of the Chemical Engineering Series, is sure to be welcomed by chemists and engineers alike.

Written by three outstanding authorities employed by the Aluminum Co. of America, the first volume treats of the discovery of aluminum, the history and development of

(*Continued on next left-hand page*)

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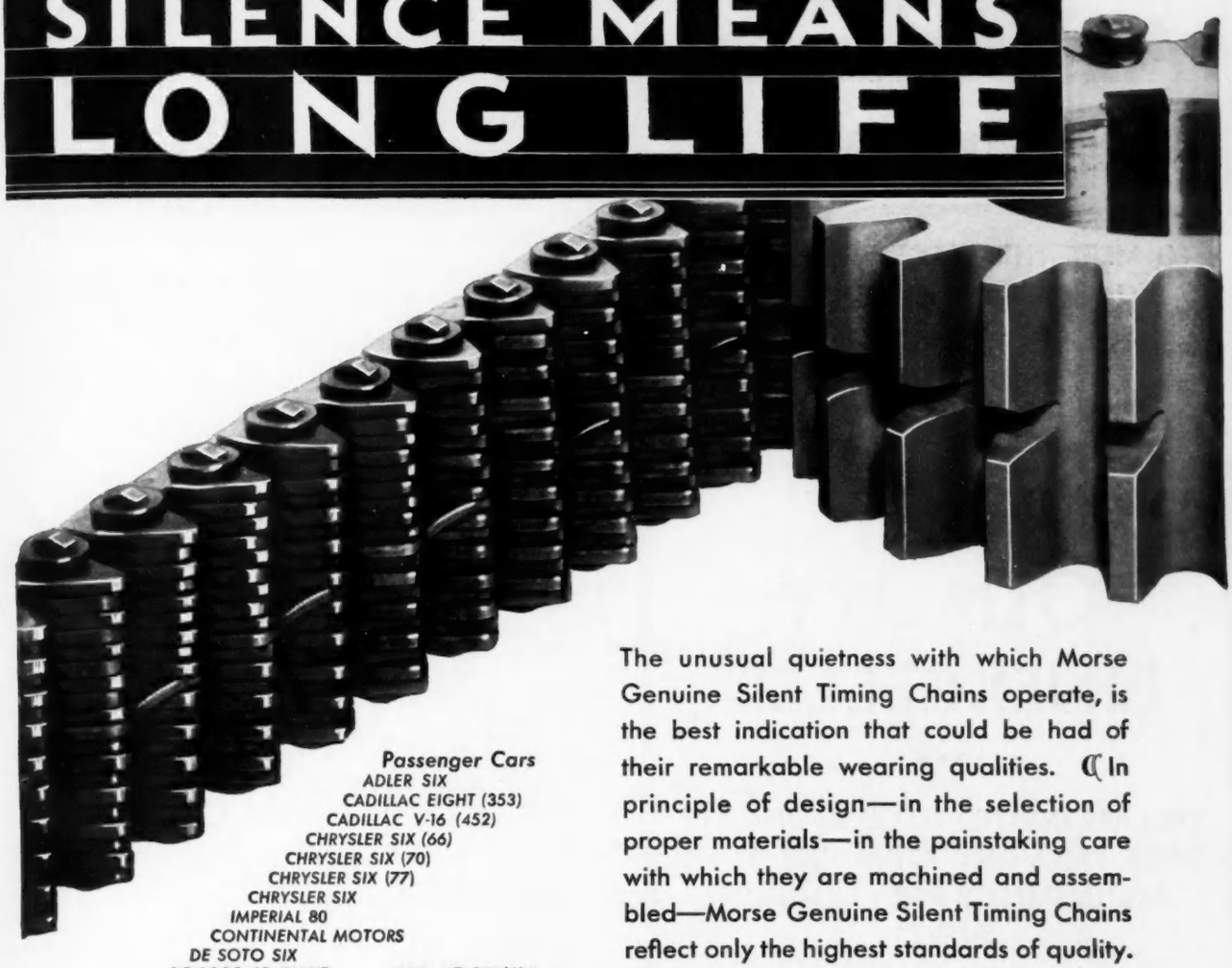
SILENCE MEANS LONG LIFE



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Notes and Reviews
Continued

the industry, the ores of aluminum, and their mining and refining for the production of pure alumina. It closes with a discussion of the production of metallic aluminum.

The second volume, which is more than twice the size of the first, deals in great detail with methods of fabrication, such as rolling, drawing and extrusion, as well as the processes of casting and machining. The authors state that the work has been written with the view of interesting not only the specialist but also the searcher for general information on the properties and uses of aluminum. Readers who are interested in the application of aluminum in special industries will find at the end of this volume a number of chapters which are concerned with their special needs. The book is one which should be in the library of every engineer having to do with aluminum fabrication.

Sparking of Steel. By E. Pitois. Translated from the French and enlarged by John D. Gat. Published by the Chemical Publishing Co., Easton, Pa., 1929, and Tokio, Japan; 89 pp., 32 illustrations. Price $2. [G-5]

The author is a graduate of the Ecole Technique and chief inspector of the Construction Section of Governmental Airships. John D. Gat, the translator, is research metallurgist of the American Sheet & Tin Plate Co.

Some experienced metallurgists can tell the composition of a steel by looking at the fractured end of the bar. Others make a rough test of its contents by putting it against an abrasive wheel and noting its characteristic spark stream. M. Pitois has attempted in this little book to reproduce by photographic record the streams of sparks emanating from the various kinds of carbon and alloy steels. One chapter is based on information furnished by John A. Houtz, spark analyst of United Alloy Steel Corp., of Canton, Ohio. There is also a chapter on the production of the spark stream in gases other than air, and one on the incrustations produced on glass by sparks formed by grinding the steel. F. L. F.

Steel-Treating Practice. By Ralph H. Sherry. Published by the McGraw-Hill Book Co., Inc., New York City and London, 1929; 388 pp. and index, illustrated. Price $4. [G-5]

In this book the author attempts to outline the requirements of practice rather than to discuss the theory of steel-treating, pointing out, however, that practice depends upon technical principles and must therefore be discussed in the light of these principles. The book has been written for those who desire a general view of the field covered in steel-treating practice, not for the technician.

The text presents simple basic principles as they are connected with practice and includes, not only standard methods, but also "variations that are important in industry and behind which the dollar sign stands prominently."

Chapters of interest especially to the automotive engineer are: Commercial Steels Composition and Working Diagrams, Physical Properties and the Stresses of Service, Inspection and Control.

The Cellulose Lacquers. By Stanley Smith, O. B. E. Ph. D. Published by Isaac Pitman & Sons, New York City, London, Toronto, and Bath, 1928, 142 pp. and index. Price $2.25. [G-5]

Ever since the Great War forced the rapid finishing of airplanes and motor-cars, the cellulose lacquers have played an important part in the automotive world. Therefore engineers who are engaged in this phase of finishing work will be interested in this "practical handbook of cellulose lacquer manufacture," written by a consulting industrial chemist who is a lecturer at the Borough Polytechnic Institute, London.

(*Continued on next left-hand page*)

A Market All Your Own

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Notes and Reviews

Continued

After discussing the nature and composition of the nitrates of cellulose, the author describes the resin, solvents, diluents and plasticizers. Other chapters are devoted to cellulose-lacquers plant and machinery, the design of formulas, pigments, and industrial application. There is also a chapter on hygrometry, because "the chief source of failure and disappointment in this new industry may be traced to the deposition of moisture on the film during the process of drying off." F. L. F.

MISCELLANEOUS

Die Maschinenelemente. By Friedrich Barth. Published by Walter de Gruyter & Co., 118 pp.; 152 illustrations. **[H-1]**

So rapid has been the progress in the technique of machine design that the literature on this phase of engineering now embraces a vast amount of complex material, the author explains, and the reader seeking to easily and quickly get at some particular piece of information finds himself confronted with the necessity of making a long search.

The present volume has been compiled to serve this class of reader. While it is not a new work, four previous editions having appeared, this last form is said to incorporate so much revision as to render it almost an original treatise. As is to be expected in such an attempt to compress a vast amount of material into the small compass of more than 100 pages, only an introduction and a review of the subject are given. Methods of calculation are merely indicated, but, it is thought, with sufficient clarity to enable the reader to complete the detail.

Industrial Economics. By Dexter S. Kimball. Published by the McGraw-Hill Book Co., New York City and London, first edition, 1929; 307 pp. and index; illustrated. Price, $3. **[H-5]**

One of the McGraw-Hill Industrial Management Series written by the dean of the College of Engineering of Cornell University, this book is an attempt to stimulate thought along lines of industrial management. It presents the idea that there may be science back of industrial management and that it may be worth while for foremen, superintendents, and higher administrative officers to inform themselves more fully concerning the economic basis of modern industry.

Tracing the history of invention and manufacturing progress, and from time to time drawing interesting conclusions regarding the effect of these changes, the author sets forth certain basic industrial principles—principles of production costs and the relation between productive factors. He discusses also the growth of inventions and enterprises, the economics of organization and the significance of standardization. He closes with a few chapters devoted to science in industry and management, measures of management, and the mechanization of industry.

Admitting that the difficulties of the problem are manifest, that management is and must remain partly personal in character, he concurs with Prof. Joseph N. Roe, who truly states: "Those elements which constitute art and are based on personal skill, leadership and imagination, as well as the vital element of integrity, can and should be left to judgment as heretofore."

Life Expectancy of Physical Property. By Edwin B. Kurtz. Published by The Ronald Press Co., New York City, 1930; 202 pp. and index, 90 figures. Price $6. **[H-5]**

Despite the successful and widespread use of the principles and methods which have been employed by life insur-

(*Continued on next left-hand page*)

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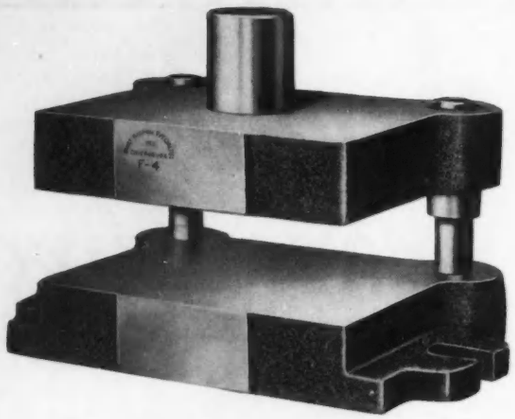

A Byword

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Carloads of standardized, interchangeable die sets are now in stock . . . ready for immediate use. They can be ordered by 'phone or telegram. You can go into production "in high" when you have the Danly factory and Danly assembly plants to fall back upon. Leading manufacturers of automobiles and accessories depend more and more each year upon Danly Service.

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Notes and Reviews

Continued

ance actuaries in dealing with the problems of human mortality tables, no attempt has been made in the past to apply these principles to determine the service life of physical property.

The author, who is head of the Department of Electrical Engineering at the State University of Iowa, has, over a period of 14 years, compiled records from which he has been able to assemble mortality tables. The life characteristics of actual physical property, covering a wide range of structures and machinery, have been calculated and analyzed, and such relations as could be readily discovered are noted.

The text is freely illustrated with charts and diagrams and it is easy to believe that the author will realize his hope that his basic findings will "in time become the foundation stones on which theories and practices of depreciation and valuation will be erected as superstructures."

Work Routing in Production. By John Younger. Published by The Ronald Press Co., New York City, 1930; 109 pp. and index, illustrated. Price $2.50. **[H-5]**

The author of this book needs no introduction to members of the Society. Mr. Younger has been interested for many years in the production activities of the Society and served as chairman of numerous production meetings; at present he is serving as Chairman of the Production Activity Committee and Vice-President representing Production.

In this book Mr. Younger points out that after a production plan has been set up there are, in general, three steps in making it effective: work routing, scheduling, and dispatching. He devotes his text solely to the development of the underlying principles of these parts of the production program and to the practice and methods whereby they are made effective.

A selected bibliography is appended.

MOTORCOACH

Das Trolleybus-System, ein Neuzeitliches Verkehrsmittel. By L. Betz. Published by M. Krayn, Berlin, Germany; 158 pp.; 96 illustrations. **[J-1]**

Hailing the trolley-bus, perhaps more commonly known in this Country as the trackless trolley, as the modern mass-transportation means *par excellence*, the author devotes his book to reintroducing this hybrid vehicle to the land of its origin. Although this transportation agency, which derives its power from an overhead electric cable but does not run on tracks and therefore, so far as ground contact is concerned, is flexible within the limits imposed by its cable connection, was first conceived of in Germany, its development has been furthered far more energetically in other countries, notably England. The author's concern is to bring to his German readers the facts regarding these developments and to indicate their significance for Germany.

The firm of Siemens & Halske originated the trackless trolley, according to the author, and in the story that he gives of its development an experimental vehicle tested in 1882 is pictured as the first incorporation of the idea. The past and present use of the vehicle in England, France and the United States, is reviewed. References to its use in this Country include only its operation on Staten Island and in Salt Lake City and the product of the Atlas Truck Co.

The technical aspects of the subject are covered by a thorough description of the electrical equipment and the design and construction of the vehicle in Germany and England. Its operation in England is covered by a description and statistics, and finally the author points out the opportunities for utilizing its services in Germany.

(*Concluded on next left-hand page*)

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HE'S TOUGH—but he has a tough job. Because he has handled that job well he wears Chevrons—the Army's reward for long and faithful service.

In the industrial field Garlock Chevron Packing handles tough jobs well. It also renders long and faithful service. Like the Sergeant, it has won its Chevrons.

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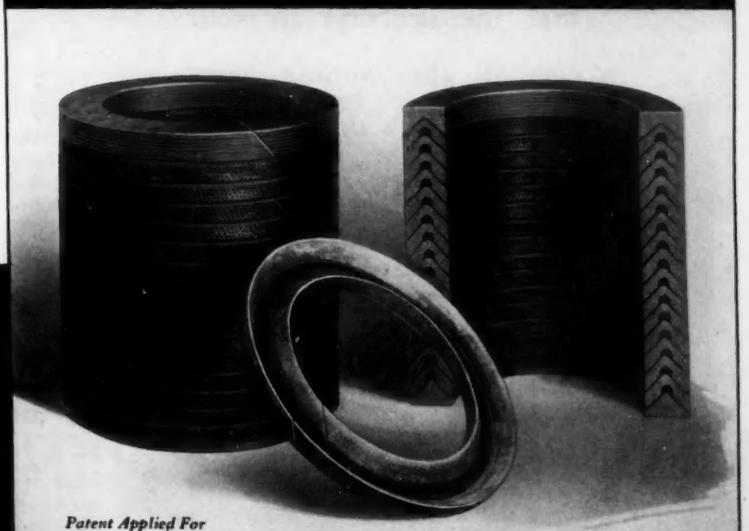
GARLOCK-430 ... *adjusts itself automatically*



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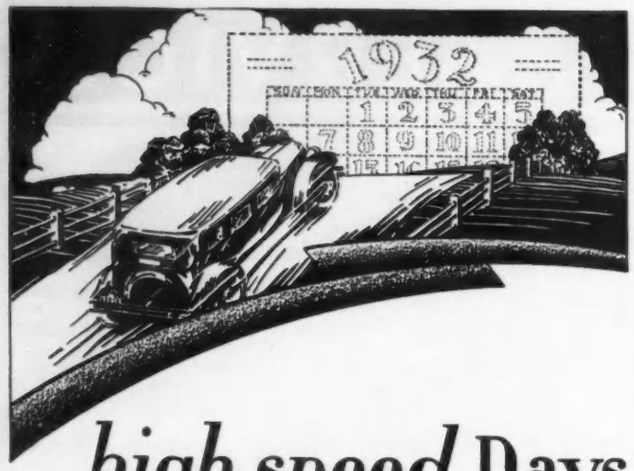

high speed Days

The cars you are building today must be built for tomorrow. The motors of six years ago are as obsolete as grandfather's watch. Only modern engineering by forward looking engineers can meet the speed and traffic requirements of the future. B. C. A. engineers are planning and looking ahead so that the bearings in your cars in the coming years may be up to the demands of the public.

Notes and Reviews

Concluded

PASSENGER CAR

L'Automobile et la Traction Avant. By J.-A. Grégoire. Published in *La Technique Automobile et Aérienne*, second quarter, 1930, p. 46. [L-1]

Citing the proverb as to the stupidity of placing the cart before the horse, the author predicts that in a few years the front-wheel drive will have won a complete victory in the field of automotive design. Logic, he says, is on the side of this type of construction, and industry eventually yields to logic. His long and favorable experience with front-wheel-drive cars is adduced as a practical argument to support the theory advanced.

Failure to develop a suitable universal-joint has for 30 years diverted automotive design from its proper channels, in the author's opinion. He goes into the theory of universal-joint design and describes four early types used in front-wheel-drive cars; the Crompton, patented in England in 1907; the Retal, used on the electric vehicle A. E. M.; the Mechanic, entering into the design of the German Voran and the Chavrier. In his judgment only three joints successfully transmit uniform motion over varying angles: the Weiss, the Parvillé and the Tracta. He describes in detail the Tracta joint, which he believes will find most favor because (*a*) it transmits uniform motion over even wide angles; (*b*) its construction, consisting of only four parts, is simple and hence cheap; (*c*) its lubrication is complete; and (*d*) its wear is virtually nil.

La Tracta. By René Charles-Faroux. Published in *La Vie Automobile*, April 25, 1930, p. 139. [L-1]

Considerable space is given, in this description of the front-wheel-drive Tracta passenger-car, to the universal-joint. This, the author claims, is simple in construction, consisting of only four pieces; is easily mounted, calling for the setting of only four screws; is well lubricated, being virtually enclosed in a sphere filled with oil; and can transmit movement from one shaft to another between which there is a considerable angle.

The front wheels are independently sprung, and the mechanism of the four-wheel brakes, so far as the front wheels are concerned, is arranged so as to be part of the suspended weight of the vehicle. Shock-absorbers are provided for all four wheels. Other interesting details are mentioned and the performance of the car in the 24-hr. endurance test at Mans is cited as proof of its capabilities.

TRACTOR

The Diesel-Engine Tractor and Excavator. By C. G. A. Rosen. Paper presented at the Third National Oil and Gas Power Meeting, State College, Pa., June 12 to 14, 1930. [M-1]

A general picture of the problems involved in the development of a Diesel engine for track-type tractors is presented in this paper.

The author reviews an investigation conducted by the Caterpillar Tractor Co. three years ago, which included the examination of the oil-engine field at that time, and the observation under field operation of a representative group of high-speed engines. At the conclusion of these tests it was decided that no available Diesel engine could be substituted for the established gasoline engine and meet the exacting service demands. Therefore it was thought advisable to experiment with fuel systems and to organize a program of investigation for the development of an oil-burning engine of satisfactory combustion characteristics. Details of the latter tests are given in the paper. The results reveal the weaknesses of each fuel system and indicate the path of development for a more efficient method of burning fuel, incorporating a control of practical operating quality.

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Capacity 2500 Tons Per Month

Any Type—Any Size—Up to 500 Lbs.

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Personal Notes of the Members

(*Continued from p. 243*)

Knut Henrichsen, former assistant chief designer for the Curtiss Aeroplane & Motor Co., of Garden City, N. Y., recently assumed the position of project designer with the Curtiss Robertson Mfg. Co., of Robertson, Mo.

L. F. Hosley, until recently general manager of Du Pont Motors, Inc., is now general manager of the Indian Motorcycle Co., of Springfield, Mass.

Grosvenor Hotchkiss, who had been serving the Western Union Telegraph Co. in New York City as engineer, recently formed a connection with the Teleregister Corp., also of New York City.

Scott F. Hunt, a former sales engineer for the Bendix Stromberg Carburetor Co., of South Bend, Ind., has been made assistant chief engineer of that company.

Dale P. Jacobus is now a partner in the Jacobus Engineering Co., of Cleveland. His previous connection was with the Foster Vernay Corp., of Brooklyn, N. Y., which he served as research engineer in the new development department.

Abbott A. Lane, formerly an engineer with Allied Engineers, Inc., has accepted a position in the research department of the Gulf Companies, of Pittsburgh, and assumed his duties on July 1.

Carlton J. Lauer is now chief engineer in the refrigeration division of Brooks Steam Motors, Inc., of Buffalo. He was previously a member of the engineering department of Chrysler Motors, Inc., of Detroit.

Harvey J. Lidkea recently became the owner and operator of a service station in Detroit. Before that he was a sales engineer in the railway department of the Timken-Detroit Axle Co., of Detroit.

P. B. MacEwen, a chassis assembly engineer for General Motors of Canada, Ltd., of Oshawa, Canada, has been transferred from Oshawa and is now a resident engineer of the company at Regina, Saskatchewan, Canada.

William J. Mayer recently assumed the duties of sales engineer with the Brockway-Indiana Motor Truck Corp., in New York City. Previous to accepting this position he was assistant manager of the associated products division of the J. G. Brill Co., of Philadelphia.

Robert S. Merithew, a former student engineer with the H. H. Franklin Mfg. Co., of Syracuse, relinquished that post recently to accept a position with the Eclipse Aviation Corp., of East Orange, N. J.

Ernest J. Opie has become general superintendent of the Girard Model Works, Inc., of Girard, Pa. Previous to accepting this position he was chief engineer and plant manager for the Bailey Burruss Mfg. Co., of Atlanta, Ga.

F. M. Say announces that he is now base maintenance manager for the New York, Rio & Buenos Aires Line, Inc., and is resident in Port-of-Spain, Trinidad, British West Indies. His previous connection was as service manager for the Central Air Terminal Co., of Chicago.

William E. Schroeder has resigned his post as sales engineer for the Haskelite Mfg. Corp., of Chicago, and is now assistant to the technical manager of the Vacuum Oil Co., in Chicago.

John M. Sickler is now engaged in laboratory research work at the Cleveland plant of the Midland Steel Products Co. He was formerly research assistant at Purdue University, West Lafayette, Ind.

William Alexander Simms, until lately junior engineer with the Western Electric Co., at Kearny, N. J., is now field manager in charge of drilling and production for the Simms Oil Co., of Dayton, Ohio.

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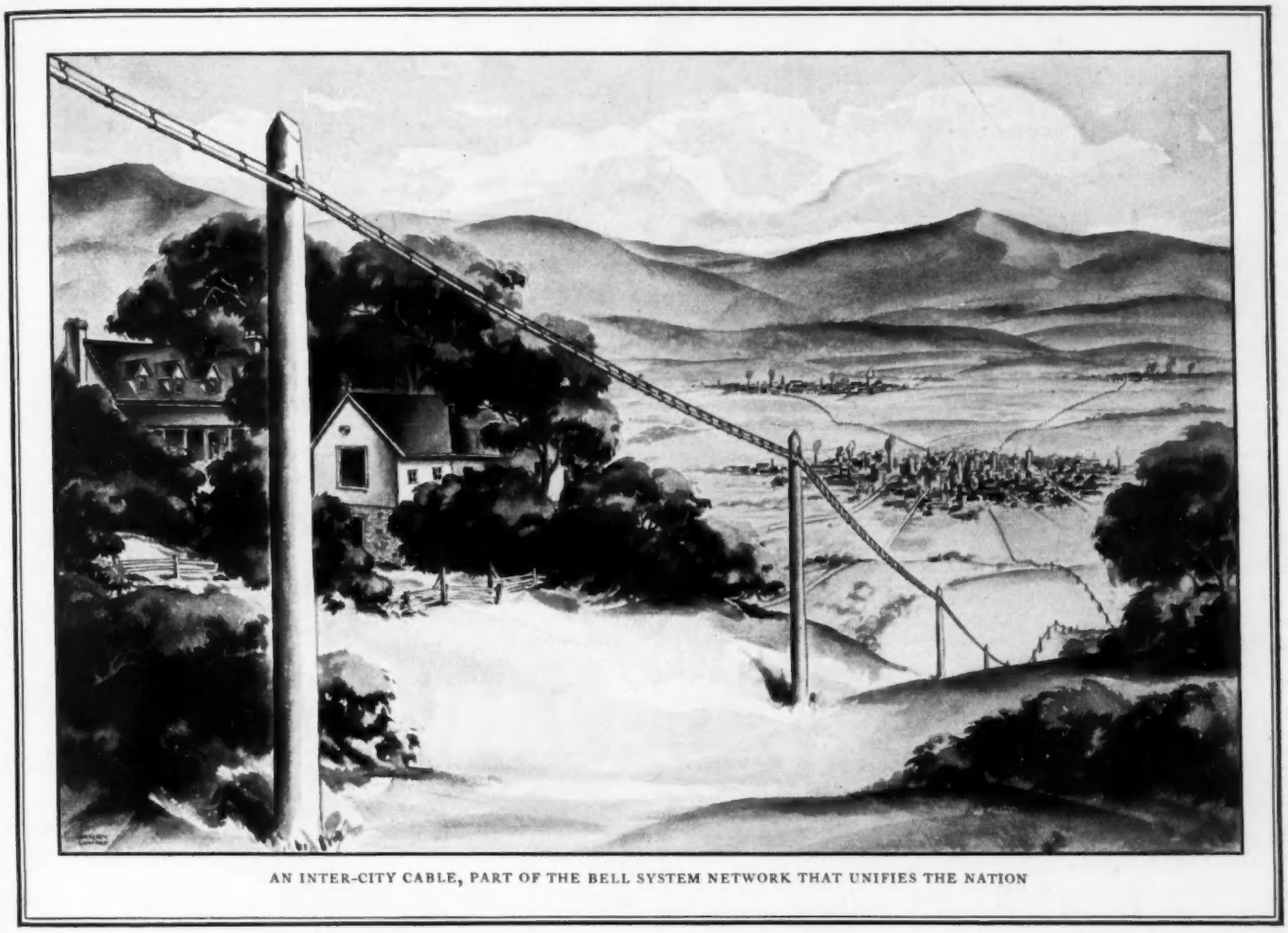

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In order to meet the telephone needs of the country most effectively, the operation of the Bell System is carried on by 24 Associated Companies, each attuned to the area it serves. Working with these companies is the staff of the American Telephone and Telegraph Company, giving them the benefit of its development of better methods.

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A low-pressure gauge that will withstand very *high* pressures.

Rugged, strong. Built to take punishment —shocks, vibration, overloads—and come back for more.

Its splendid design and construction — with *restraining back-plate* which prevents loss of accuracy of calibration under excessive pressures— is the feature that sets it distinctly apart.

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Over 7,000,000 Sold.

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Personal Notes of the Members
Concluded

Leroy P. Sterling, formerly special investigator in the laboratories of the Lycoming Mfg. Co., of Williamsport, Pa., has severed his connection with that company. His plans for the future have not been announced.

William M. Stilger has been promoted from the position of manager of parts service to general service manager of the Hudson Motor Co. of Illinois, at Chicago.

Earl A. Taylor, who has been serving as manager of the Yellow Sleeve Valve Engine Works, Inc., of East Moline, Ill., is now factory manager in charge of plant No. 1, of the General Motors Truck Co., at Pontiac, Mich.

Conrad A. Teichert, formerly employed as engine designer and mathematician for the International Harvester Co.'s Fort Wayne, Ind., plant, is now employed in the research division of the Bendix Brake Co., of South Bend, Ind.

A. J. M. Tuck, until lately sales manager of the Amaloy division of the American Machine & Foundry Co., of New York City, is now director and sales division manager for that company.

Kenneth W. Warren has severed his connection as chief design engineer with the Le Blond Aircraft Engine Corp., of Cincinnati, and announces that he has accepted a position as research engineer with the Bendix Aviation Corp., of South Bend, Ind.

John A. Watts, formerly an engineer with the Western Electric Co., of Chicago, is now occupying a similar position with the Standard Oil Co. (Indiana) and is stationed at Des Moines, Iowa.

Wayne H. Worthington recently became a research engineer with the John Deere Tractor Co., of Waterloo, Iowa. He was previously chief engineer with the Gleaner Combine Harvester Corp., of Independence, Mo.

Employment Service

The Society endeavors to serve its members as completely as possible in the matter of employment. Approximately 150 men are placed in positions each year through its Employment Service, at salaries ranging from $2,500 to $10,000 or more per year.

Through a system of BULLETINS mailed twice each week, addressed for the personal attention of the General Managers, the Society keeps in touch with approximately 1000 companies of good standing in the industry.

Contact is maintained also with Technical Schools and Publishers of Trade Magazines.

BULLETINS OF POSITIONS AVAILABLE are mailed frequently to members to whom they are of interest.

The Bulletins of Men and Positions Available are mailed regularly to the Secretaries of the Sections of the Society for the purpose of making the Bulletins readily accessible to companies and to members located near the headquarters of the respective Sections.

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